OFFICIAL
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ASHRAE

JOURNAL

Heating • Refrigerating • Air Conditioning • Ventilating

AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC.

ASHRAE CAMPAIGN STARTED FOR UNITED ENGINEERING CENTER

The UEC is being erected on New York's United Nations Plaza.

We, of ASHRAE, have an important role to play in the operation and construction of the Center.

Our headquarters will be combined and occupy over 4,000 sq ft in the building when it is opened in the spring of 1961.

The participating societies have had their part in the ten million dollar construction fund campaign underway for some time . . . the goal for the ASHRAE campaign, just starting, is two hundred and fifty thousand dollars.

Members are counted upon to give according to their abilities, which vary according to those rewards which engineering has afforded them.

—MERRILL BLANKIN, Chairman
ASHRAE-UEC Fund-Raising Committee

See pages 54-55-56

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Comment

SO WHAT ELSE IS NEW?

Sometimes it seems as though the coming of Fall, with vacations past and plans firming for the relatively uninterrupted and intense months ahead, is more of a New Year than comes with the actual flipping of the calendar in mid-Winter.

In any event, the Editors of your ASHRAE JOURNAL have that feeling this year. Concomitant with various post-merger plans, your JOURNAL will undergo certain evolutionary changes that should adapt it better to the overall interests of ASHRAE members.

This Society, any society, exists for the furtherance of the interests of its members and for the promotion of the general public welfare. The only question is when is or is not that goal achieved. You can assist by letting your officers know, as President Arthur J. Hess urged in the President's Page of our May issue.

This is your ASHRAE. Where it heads, where it serves, what it means is, in the larger sense, not the determination of officers or of staff but of you, the members.

OFF WITH THE OLD; ON WITH THE NEW

It is difficult to look forward without looking back. Partly, because every future must be based upon at least some of the lessons of past experience. Partly, because even the most eager seeker of new worlds must have his twinges of nostalgia.

But a few days since, ground was broken for the new United Engineering Center in New York which will house a predominant number of national engineering societies, including ASHRAE, and offer room for foreseeable future expansion. It was an inspiring and noteworthy moment.

Fund-raising contributions from all participating organizations will presently aggregate $10 million. Members of one society have over-subscribed their quota; others with campaigns just getting under way, as with ASHRAE, have this accomplishment as yet unrecorded.

But there comes the inevitable backward look toward that earlier stimulus to engineering progress made by Andrew Carnegie when he initiated the movement that brought about the original Engineering Societies Building in New York. We revert, perhaps a bit fondly, to some of the astonishingly important milestones in engineering that were foretold and later reported upon at meetings there and to the programs and cooperative undertakings which achieved definition within its walls.

It scarcely belittles Tomorrow to recognize the merits of Yesterday.

Edward R Searles
Editor

Engineering the temperature...

IN SINGAPORE

Twelve Recold Multizone units air condition the American International Assurance Building. These Recold Multizone units, one on each floor, supply 108,000 CFM of conditioned air for this modern, 34 zone, office building. One standard Recold air conditioning unit cools the executive penthouse.

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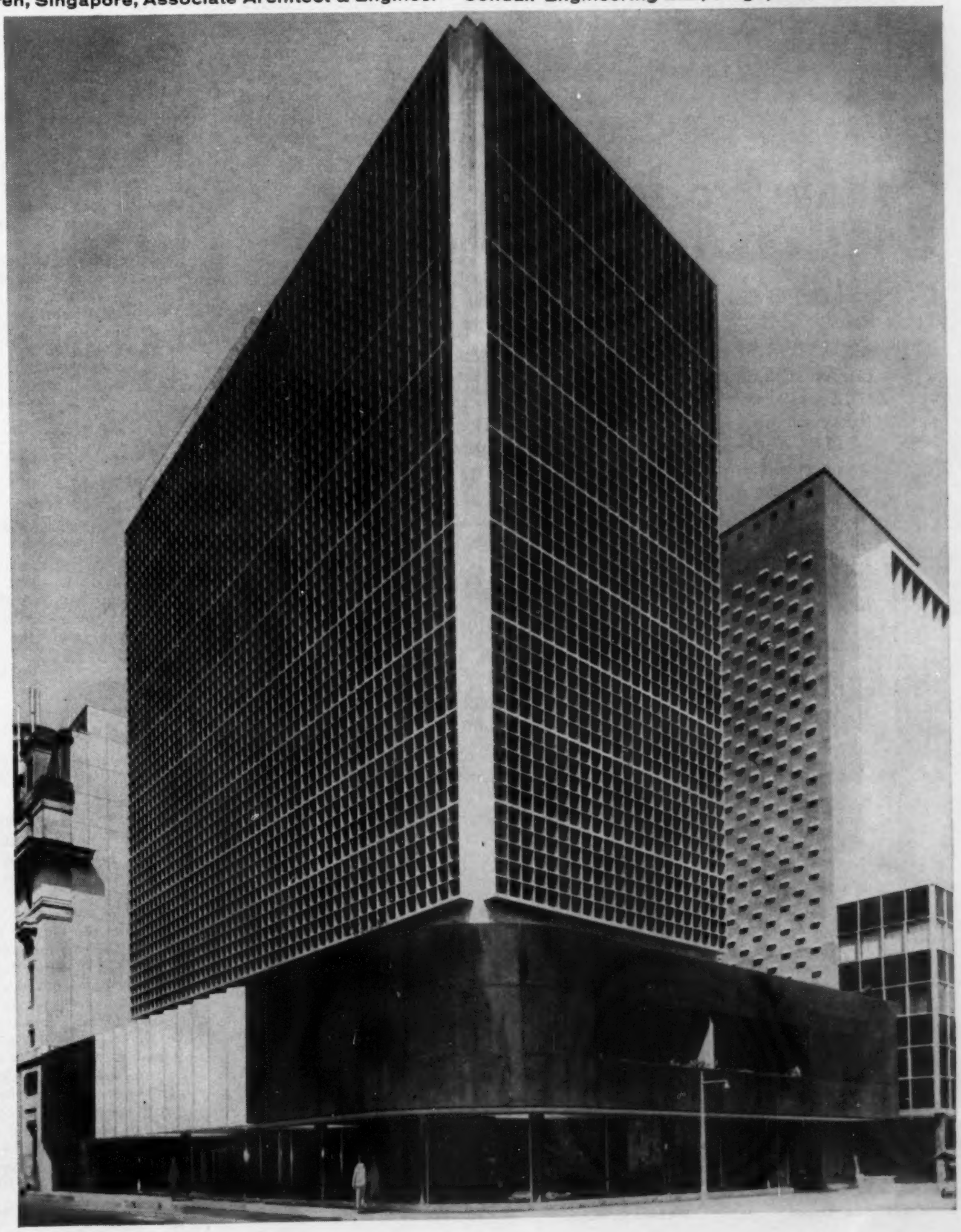

Now that you mention IT—

EXCEPT ENTHALPY

To the Editor:

Author D. D. Wile presented a very informative history of psychrometric charts in his "Psychrometric Charts Past and Present" in the August issue of the JOURNAL.

I am inclined to agree with his conclusions regarding chart construction, except those referring to enthalpy. Perhaps because of daily use, I find no confusion between wet-bulb and enthalpy lines, and I do object to making the correction calculations necessary with "enthalpy deviation curves" or similar devices.

I agree fully that "percent saturation" curves serve no useful purpose and could be omitted.

As to high altitude charts, my particular interest, I have prepared charts for 23, 25 and 26 in. Hg. atmospheric pressures adequate to cover altitudes to 8000 ft without serious error.

ROGER W. HAINES

Bridgers and Paxton
Consulting Engineers
Albuquerque, N. M.

To the Editor:

Figs. 1, 7 and 10 of my "Psychrometric Charts — Past and Present" in the August issue of the JOURNAL were printed on their sides, rather than in vertical positions. Unfortunately, the position is not immediately evident and thus misrepresents the intent of these figures.

D. D. WILE

Recold Corporation
Los Angeles, Calif.

CARBON-RING COMPRESSORS?

To the Editor:

Norman Sharpe's "Experimental Research on Lubrication in Refrigerant-12 Systems" in the August 1959 issue emphasizes the problem of oil in refrigerant lines.

Oil is a continuing problem, starting at design stages and continuing through the maintenance life of the system. My question is: Why not use oil-free compressors (carbon-ring, etc.) such as are used for instrument air in process plants? They are more expensive, today. But certainly were they adopted as an industry standard manufacturing costs would decrease.

This question has been ducked by several compressor people in the last couple of years. I would be interested to know if any study has been made of this and what the results may be.

OLEG N. DUDKIN

United Engineers & Constructors Inc.
Philadelphia, Pa.

Late news highlights

Enrollment trends Engineering Manpower Commission's recent study of enrollment trends in Technical Institutes which offer curricula accredited by Engineers Council for Professional Development revealed the following data: 1958 first-year enrollments increased over 1957 by 1.2% (however, the increase in 1957 over 1956 was 10.2%); less than one-third of the institutes reported receiving a lower number of applications from qualified candidates; less than 30% of the institutes accepted a smaller number of applicants for 1958 enrollments; and overall first year enrollment is expected to increase in the Fall of 1959.

Preserving food *Technology of Food Preservation* by Norman W. Desrosier, professor of food technology, Purdue University, covers comprehensively the technology and fundamental principles of food preservation. Included are chapters on refrigerated storage of perishable commodities; principles of food freezing; and preservation of food with ionizing radiations. The book, which contains 418 pages and 180 illustrations, is published by the AVI Publishing Company, P. O. Box 388, Westport, Conn., price is $8.50 (foreign $9.50).

Future methods? One of the major topics to be discussed at the Building Research Institute's Fall Conferences, November 17-19, in Washington, D. C., will be New Methods of Heating Buildings. John Everetts, Jr., ASHRAE fourth vice president, is chairman of the full-day program on the opening day of the Conferences which is expected to bring out developments that may have considerable impact on buildings planned for the 1960's. New air heating methods, new wet heating systems and new radiant heating systems will be presented by researchers in these fields, as well as discussion of new electric heating systems including direct resistance heating, heat pumps and thermoelectric systems. Additionally, data on progress made in solar heating will be given.

Two instead of one Installation of two oil burners in a single boiler is gaining favor with plant operating engineers in a number of institutions and commercial and industrial establishments, according to the Oil Heat Institute of America, which lists applications where two small burners are cited as doing a better overall job than one large one. Furthermore, the Institute reports, two smaller burners may provide an added safeguard in the event of a burner failure; smaller burners usually have standard parts; and they are easier to service.

Freezing corn University of Minnesota's Institute of Agriculture recommends these two steps for freezing sweet corn – speed from garden and market to freezer and scalding corn the correct time. Information on freezing is given in the publication, "Freezing Fruits and Vegetables," Extension Folder 156, available from Bulletin Room, Institute of Agriculture, University of Minnesota, St. Paul 1, Minn.

Basic manual Design and installation of heating, ventilating, air conditioning and dehumidifying systems are covered in the manual, *Basic Mechanical Engineering*, available from the U. S. Navy Bureau of Yards and Docks. The 106-page manual, PB 151693, may be ordered from the Office of Technical Services, U. S. Department of Commerce, Washington 25, D. C., for $2.50.

Thermoelectric tests An experimental three-purpose device, combining into a single system a thermoelectric air conditioner, space heater and refrigerator-freezer, to be built for the Navy by Westinghouse Electric Corporation, will be used to test the suitability of thermoelectricity for air conditioning and refrigeration on ships. The three thermoelectric elements can be individually removed and replaced and can act as building blocks for a thermoelectric system of larger size. According to Westinghouse, the air conditioning portion will have a full one-ton capacity. The three components will be so designed that they can operate independently of one another.

ARI Conference President Harold J. Humphrey of the National Association of Frozen Food Packers, Arthur S. Goldman, Market Research Director of *House & Home* magazine, Joseph Rorick, Assistant for Planning and Construction, International Business Machines Corporation, Inc. and H. E. Ziel, associate of Albert Kahn Architects and Engineers, will discuss various types of applications of mechanical cooling and the need for cooperation between the industry and users of its products at the ARI Conference Session, a special feature of the 11th Exposition of the Air Conditioning and Refrigeration Industry, on November 3 in Atlantic City, N. J. The conference is planned to highlight the usefulness of applications of the refrigeration cycle in both refrigeration and air conditioning.

Detectors chilled An 8-oz refrigerator, which can be used to chill airborne infrared target detectors down to –350 F to make them ultra-sensitive and therefore capable of spotting aircraft or missiles from greater distances than is now possible, has been announced. An application of a new process for producing extremely low temperatures, developed by Dr. Howard O. McMahon, vice president and William E. Gifford of Arthur D. Little, Inc., the miniature refrigerator used in infrared detectors could be used in a wide variety of industrial and military applications.

Soldering manual Virtually all phases of soldering are described in the manual published by the American Welding Society, which also lists chemical composition of numerous solders, together with flux formulations for various metals. Copies of the 176-page manual, which contains 81 illustrations and 34 tables, may be obtained from AWS, Department T, 33 West 39th Street, New York 18, N. Y., for $5.

Cryogenic Conference More than 500 scientists and engineers attended the Fifth Annual Cryogenic Engineering Conference at the University of California at Berkeley, Calif., September 2-4. A review of work currently undertaken by the National Bureau of Standards included reports on NBS liquid hydrogen investigations, conducted by the Bureau's Cryogenic Engineering Laboratory, relating to storage, transportation and correlation and compilation of thermodynamic data for hydrogen; and the new CEL Cryogenic Information and Data Center, which indexes and stores cryogenic technical literature.

Safety award National Safety Council's Association Safety Award for 1959 has been presented to the National Association of Refrigerated Warehouses in recognition of a substantial reduction in the accident frequency rate of its members over the past five years. According to NARW Safety Committee Chairman A. Oakley, Jr., the 1958 accident frequency rate of 24.9 was the lowest on record for the industry. The Award, made annually on the basis of improved safety experience and new or expanded safety activities, was won previously by NARW in 1955.

RSES meeting Among the subjects to be covered at the 23rd annual meeting of the Refrigeration Service Engineers Society, to be held October 30 through November 2 in Atlantic City, N. J., are: recent advances in hydronic heating and cooling; automotive air conditioning; oil migration in Refrigerants-12 and 22 systems; and thermoelectric refrigeration.

Britain to exhibit Part of a 17-day British Exhibition of industry, technology, science and culture, to be held in New York, N. Y., June 10-26, 1960, will be a display of British air conditioning, featuring the most recent equipment of leading firms in the field.

Frozen poultry Effects of frozen storage at 10 F, 1 F and –10 F for a period of 18 months on the breast and thigh of broilers, studied by means of organoleptic and physical tests, are reported in Research Bulletin 835 of the Ohio Agricultural Experiment Station, Wooster, Ohio, "Nutritive Value and Consumer Acceptance of Frozen Poultry" by Inez Prudent.

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How refrigerant properties affect
Impeller dimensions

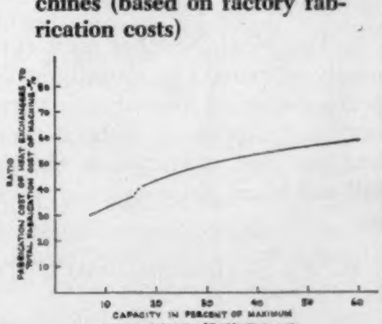
F. J. WIESNER, JR.

H. E. CASWELL
Member ASHRAE

Centrifugal compressors for refrigeration, especially the chilling of large volumes of water for use in relatively large air conditioning systems of the central station type, have gained broad acceptance. The original concept (that of Dr. Willis H. Carrier, 1921) has been developed and expanded, with a multitude of improvements, to the point where refrigeration machines using centrifugal compressors are big business. The growth rate is substantial, in spite of the more recent development of such other means for chilling large quantities of water as the lithium bromide absorption refrigeration machine.

The use of centrifugal compressors in refrigeration machines has paralleled and hinged upon the simultaneous development of other machine components, and has been influenced especially by the availability of suitable refrigerants. The heat exchangers used in association with a compressor to create a complete refrigeration machine (evaporator or cooler and condenser) are the major components so far as bulk and weight are concerned, and show high in the overall machine cost. Fig. 1, covering one line of hermetic centrifugal water chilling machines, indicates that the factory fabricated cost of heat exchangers typically varies from 30% to as much as 60% or more of the total fabricated cost (excluding the motor parts), depending upon the size of the machine and the design repercussions of the refrigerant used. Thus, the heat exchangers deserve as much attention from the designer as the compressor, and the design of all components is obviously inter-related. The objective must be to unify an optimal combination of components for low "first" cost, or for minimum customer "owning and operating" cost, depending upon the type of market. In general, low "first" cost dominates for smaller machines; "owning and operating" cost for larger units.

Fig. 1 Relative cost of heat exchangers for hermetic centrifugal water chilling machines (based on factory fabrication costs)



For centrifugal compressors, refrigerants should have acceptable cycle efficiencies, fairly large specific volumes, and reasonable "head" or "lift" characteristics. If they are in line as to cost, are non-toxic, will not ignite and are chemically stable, so much the better. In fact, the latter characteristics are virtually mandatory for acceptability in comfort air conditioning, today. A family of fluorinated hydrocarbon compounds, developed since 1921, meets the basic requirements well, yet provides considerable flexibility as to refrigeration capacity. Pertinent data for the more commonly used of these compounds are summarized in Table I. Many good refrigerants, commonly used in the past, fail to match the inherent performance characteristics of centrifugal compressors. Ammonia, for example, for a water chilling temperature "lift," requires a pumping "head" close to 60,000 ft. This means expensive multiple centrifugal staging, which can be avoided by using the new compounds described above.

This discussion concentrates on the effect of refrigerant choice upon centrifugal stage design, with a discussion of the various factors, including refrigerants, which influence and limit the optimum design. It will be limited to practical designs in the air conditioning range, and consider only those

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Frank J. Wiesner, Jr., is Senior Engineer; Howard E. Caswell is Manager, both of the Centrifugal Refrigeration Engineering Dept, Carrier Corporation. This paper was presented as "Effects of Refrigerant Properties on Impeller Dimensions and Stage Performance" at the ASHRAE annual meeting, Lake Placid, N. Y., June 22-24, 1959.

TABLE I
COMMON FLUORINATED HYDROCARBON REFRIGERANTS USED IN CENTRIFUGAL REFRIGERATION MACHINES

Formula	Chemical Name	Trade Names	ASRE Refrig. No.	Sat. Press. at 34F psia	Sat. Press. at 104F psia	Sp. Vol. Vapor at 34F Sat. ft³/lb	Sonic Velocity Vapor at 34F Sat. ft/sec
CCl_3F	Trichloromonofluoromethane	Carrene 11, Freon 11, Genetron 11	11	6.112	25.32	6.205	441.5
CCl_2F_2	Dichlorodifluoromethane	Freon 12, Genetron 12	12	46.47	139.59	0.855	453.0
CCl_2F-$CClF_2$	Trichlorotrifluoroethane	Carrene 113, Freon 113, Genetron 113	113	2.264	11.35	12.39	372.0
$CClF_2$-$CClF_2$	Dichlorotetrafluoroethane	Freon 114, Genetron 114a	114	13.38	49.48	2.236	382.0
$CHClF_2$	Monochlorodifluoromethane	Freon 22, Genetron 22	22	75.21	224.5	0.729	538.0
	Azeotropic Mixture of CCl_2F_2 and CH_3CHF_2	Carrene 500	500	54.71	165.5	0.889	480.0

fluorinated hydrocarbon refrigerants which are most commonly used in centrifugal machines. To maintain an acceptable overall machine performance, compressor performance deficiencies must be compensated by adding heat exchanger surface – which costs money. Putting it another way, a substantial improvement in the compressor (usually at little or no cost) may make it possible to reduce appreciably the associated heat exchanger surface needed. Unfortunately, compensation by increasing heat exchanger surface follows the law of diminishing returns, and can quickly expand overall costs beyond all reason.

Factors affecting the performance – It can be demonstrated that useful output (head or pressure rise) and required power input (gas hp) for any centrifugal compressor stage, operating with any vapor, are primarily dependent upon the following variables:

Quantity of Flow
Mach Number
Reynolds Number
Specific Heat Ratio, $k = C_p/C_v$

Useful head (or "lift") is expressed usually as foot pounds per pound of fluid flowing through the stage, and for the purposes of correlating data in a non-dimensional form, this quantity is commonly given as the Head or Pressure Coefficient:

$$\mu_p = \frac{g\,H_p}{u_2^2} \tag{1}$$

Practically, input to the gas may be expressed in many ways, but for the purposes of the development to follow, will be defined as the useful polytropic output divided by the polytropic efficiency, or non-dimensionally:

$$\mu_o = \frac{\mu_p}{\eta_p} = \frac{g\,H_p}{u_2^2\,\eta_p} \text{ or } = \frac{g\,H_o}{u_2^2} \tag{2}$$

As far as the quantity of flow is concerned, this is most commonly expressed in cfm at the inlet to the stage. There are also numerous non-dimensional forms for the flow quantity of centrifugal compressors. However, most investigators agree that a flow coefficient based upon the conditions of flow at the impeller tip will result in the most satisfactory correlation of data from a wide variety of stage designs and operating vapors; see Meldahl,[1,2]* Sheets,[6] and Stepanoff,[10] among others. For the purposes of this presentation, the following dimensionless impeller tip flow coefficient has been selected:

$$\phi = \frac{c_{m_2}}{u_2} \text{ (Derivation in App. I)} \tag{3}$$

The Mach Number most commonly employed in describing the performance of centrifugal refrigeration compressors, turbochargers, and the like, is known as the Rotational Mach Number and is given as:

$$M_o = \frac{u_2}{a_o} \text{ (Dimensionless)} \tag{4}$$

* Numbers refer to items in the Bibliography.

The Reynolds Number also most commonly used in analyses of this type is sometimes referred to as the "machine" Reynolds Number and is given as follows:

$$R_e = \frac{d_2\,u_2\,\rho_2}{12\,\mu_2} \text{ (Dimensionless)} \tag{5}$$

Examination of this group of equations reveals that the following vapor properties all exert some influence on the stage performance:

P-V-T characteristics of the vapor, which determine:
a. Head or Pressure Rise required for given temperature "lift" requirements, and
b. Density of the vapor at impeller discharge static pressure and temperature conditions.
Velocity of sound in the vapor at the stage inlet total pressure and temperature conditions.
Viscosity of the vapor at impeller tip static pressure and temperature conditions.
Specific Heat Ratio of the vapor.

The equations also indicate that the following physical dimensions (geometrical requirements) are of primary importance in connection with the stage performance:

Impeller Tip Diameter, d_2, which at a given speed of rotation, N (rpm), also determines the peripheral speed of the impeller, u_2, according to:

$$u_2 = \frac{\pi\,d_2\,N}{12 \times 60} \text{ (Or Approx. } \frac{d_2\,N}{229}\text{)} \tag{6}$$

Impeller Tip Width, b_2, which enters into the actual determination of c_{m_2}, the average radial or meridional velocity of the vapor at the impeller tip.

There are a few other secondary geometrical requirements or limitations for the stage, among which are the impeller tip blade angle and the maximum inlet relative Mach Number, which enter into the ability of the design to produce "peak" performance. These considerations will be discussed later, in connection with some specific examples.

Typical peak stage performance charts – As a result of the correlation of a large amount of test data,

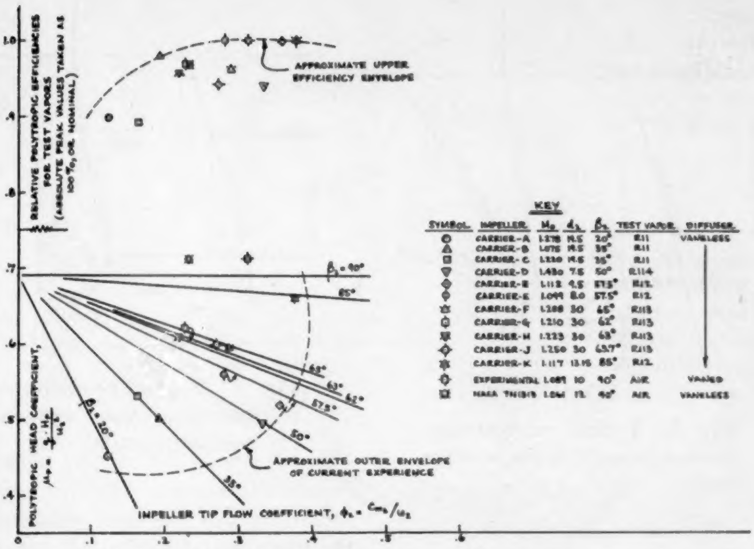
Fig. 2 Test performance correlations (at peak efficiencies) in dimensionless coordinates

similar to that shown in Fig. 2, obtained from centrifugal stages of many different designs (operating in a wide variety of vapors), taken in conjunction with other data which are available readily [principally Meldahl,[2] Sheets,[6] Balje,[7] Wosika[8] and Stepanoff[10]], the diagrams shown in Figs. 3 and 4 were constructed. These charts are felt to describe adequately the practical obtainable peak performance of any carefully designed centrifugal compressor stage, operating with any vapor. Thus, any point on Fig. 3 may be considered to be a "design point" (for no inlet pre-rotation), and it should be noted that the predominant effects of the Flow Coefficient, Impeller Tip Blade Angle, and Rotational Mach Number on both the Polytropic Head Coefficient and the Polytropic Efficiency may all be represented on a single chart. Here, it should be pointed out that the test polytropic efficiencies shown in Fig. 2 have been "normalized" and subsequently termed Relative Polytropic Efficiencies. This was accomplished by taking the absolute value of the peak polytropic efficiencies for the four best efficiency test points, (Carrier Impellers E and K, and the Experimental Impeller, all at $M_o \cong 1.10$), as a nominal (or 100%) value, and expressing all other test efficiencies in terms of this nominal.

The Relative Polytropic Efficiencies shown in Fig. 3 are also expressed in the same terms, and it can be seen that the nominal values of Figs. 2 and 3 are in agreement at a Rotational Mach Number of about 1.10 in the region of high Flow Coefficients. Fig. 4 indicates the slight (but not completely negligible) influence of Reynolds Number on the Polytropic Efficiency for several of the more common refrigerants. Note that the relative efficiencies shown in Fig. 3 are specifically labeled as representative of those attainable with Refrigerant-12, since this vapor has the highest Reynolds Number among the four most common refrigerants (11, 12, 113 and 114), used in centrifugal air conditioning machines. With reference to Fig. 4, the rest of these refrigerants would be expected to rank in descending order as follows. Refrigerants-114, 11 and 113. Of the higher tonnage refrigerants, Refrigerant-500 is expected to be equal to Refrigerant-12, while Refrigerant-22 may be considered slightly better as a result of even higher Reynolds Numbers in the air conditioning range of applications.

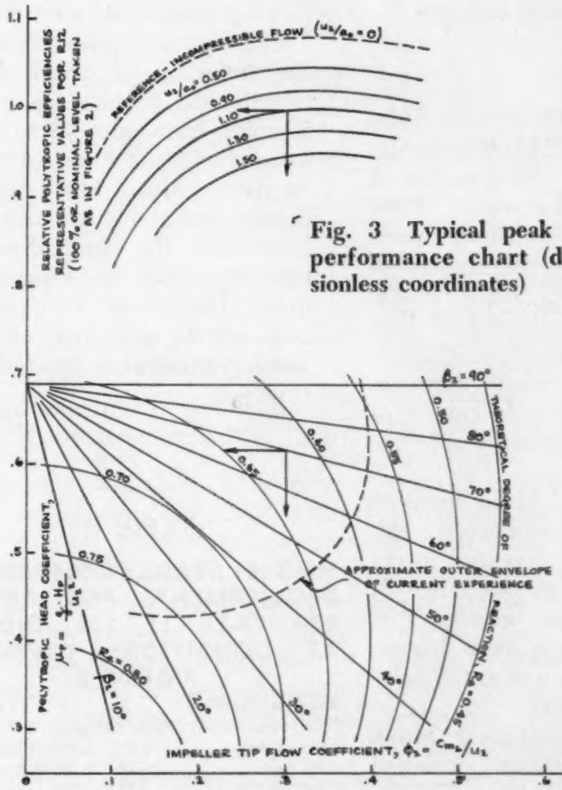
Fig. 3 Typical peak stage performance chart (dimensionless coordinates)

Again referring to Fig. 4, we have shown Relative Polytropic Efficiencies in place of the absolute values, with the reference value of the polytropic efficiency of Refrigerant-12 being chosen to coincide with that used in Figs. 2 and 3.

Further study of the Peak Stage Performance Chart, Fig. 3 results in the following conclusions with respect to the main influences of the primary variables:

For a given impeller tip blade angle, β_2 (except, of course, for radial bladed impellers), as the

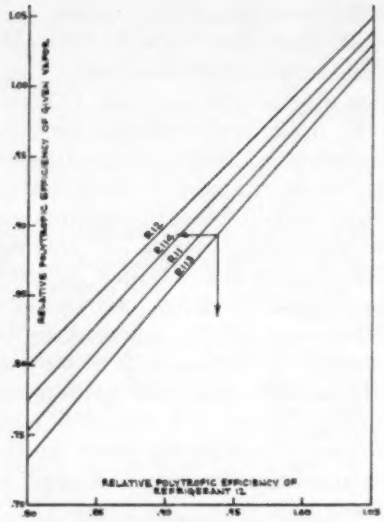

Fig. 4 Effect of Reynolds Number on the relative polytropic efficiencies of various refrigerant vapors as compared with that for Refrigerant-12

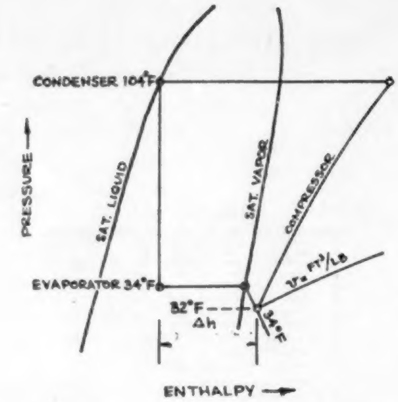

Fig. 5 Typical refrigeration cycle assumed for single stage units

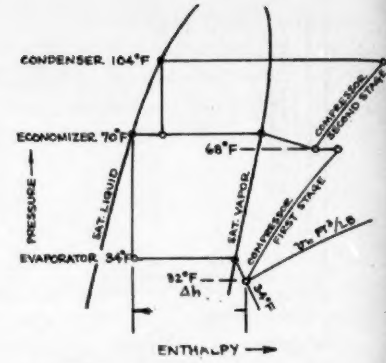

Fig. 6 Typical economizing refrigeration cycle assumed for two stage units

Flow Coefficient is increased, the Polytropic Head Coefficient decreases linearly. As a matter of interest, the theoretical Flow Coefficient at zero Head Coefficient is equal to the tangent of the impeller tip blade angle. Obviously, at a given Flow Coefficient, impellers having backward swept discharge vane angles ($\beta_2<90°$) must be run faster to produce the same Polytropic Head.

At any rotational Mach Number, maximum peak polytropic efficiencies may be expected to occur somewhere in the range of Flow Coefficients between 0.30 and 0.35 (perhaps even higher, although evidence is not conclusive in this regard).

At constant rotational Mach Number, as the Flow Coefficient is reduced appreciably below the value producing the best peak efficiency, peak design efficiencies can be expected to drop sharply. At Flow Coefficients much above the value producing best peak efficiency, we may again find that peak design efficiencies tend to drop. Although there is not sufficient experimental evidence in the region of higher Flow Coefficients, we do know that these designs will normally produce stages having supersonic inlet conditions with the common refrigerants. Such designs result in peak efficiencies which may not be expected to be as good as those obtainable with subsonic or even trans-sonic inlet designs. This situation is primarily due to the influence of shock losses, which tend to increase rapidly in going from subsonic to supersonic conditions at the inlet.

As the rotational Mach Number is increased at any constant Flow Coefficient, the obtainable peak efficiency may be expected to drop. This effect, too, becomes more serious as higher rotational Mach Numbers are applied.

In order to illustrate applications of the practical trends shown in Figs. 3 and 4, the following material is presented as typical of results obtained in an analysis of the potentials of the four common refrigerants (11, 12, 113 and 114) in centrifugal stages for an assumed air conditioning duty.

TABLE II

SINGLE STAGE COMPRESSOR REQUIREMENTS FOR REFRIGERANTS-11, 12, 113 AND 114 AT CONDITIONS GIVEN IN FIGURE 5

REFRIGERANT	11	12	113	114
Polytropic Head, ft*	8770	6860	7190	6030
Lb/min/ton	2.970	4.095	3.715	4.660
Cfm/ton	19.40	3.65	48.50	10.90
Cfm at 300 tons	5820	1096	14560	3270

REQUIREMENTS FOR FIRST STAGE OF TWO STAGE COMPRESSOR (WITH ECONOMIZER) FOR REFRIGERANTS-11, 12, 113 AND 114 AT CONDITIONS GIVEN IN FIGURE 6.

REFRIGERANT	11	12	113	114
First Stage Polytropic Head, ft*	4530	3600	3760	3200
Avg lb/min/ton (incl Economizer effect)	2.825	3.785	3.470	4.255
Cfm/ton	17.56	3.14	42.60	9.15
Cfm at 300 tons	5260	942	12800	2745

* At assumed 75% polytropic efficiency, but variations in this value with final efficiencies implied in Tables III, IV, V and VI are quite small.

Determination of speeds, dimensions and relative power requirements for a typical air conditioning application — The usual air conditioning design requirement calls for about 44 F chilled water, with a condensing water temperature of about 85 F. Obviously, the corresponding suction and condensing temperatures (or compressor requirements) could be obtained for each refrigerant by optimum practical arrangement of the heat transfer surface in the evaporator and in the condenser. Having accomplished this, it is likely that each refrigerant would have slightly different suction and condensing temperatures, which would, of course, be a direct reflection of the heat transfer capabilities of each of the refrigerants. Even these differences could be compensated for in the overall design of the heat exchangers.

Since it is outside of the scope of this paper to consider the heat transfer aspect of the problem, we must assume some typical corresponding suction and condensing temperatures which will be held constant for all of the refrigerants to be analyzed. These temperatures are most commonly found to be about 34 F in the evaporator and about 104 F in the condenser, for the water conditions given above. The assumed suction and condensing temperatures, together with the

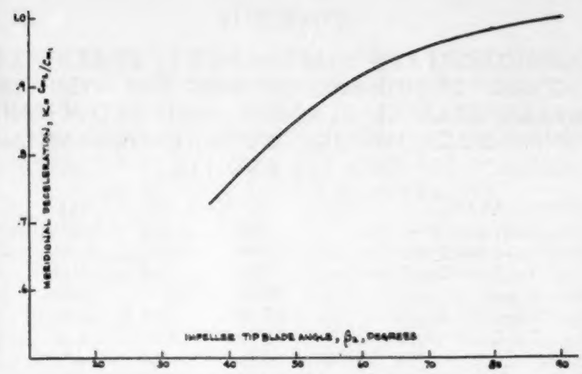

Fig. 7 Approximate allowable meridional deceleration vs impeller tip blade angle

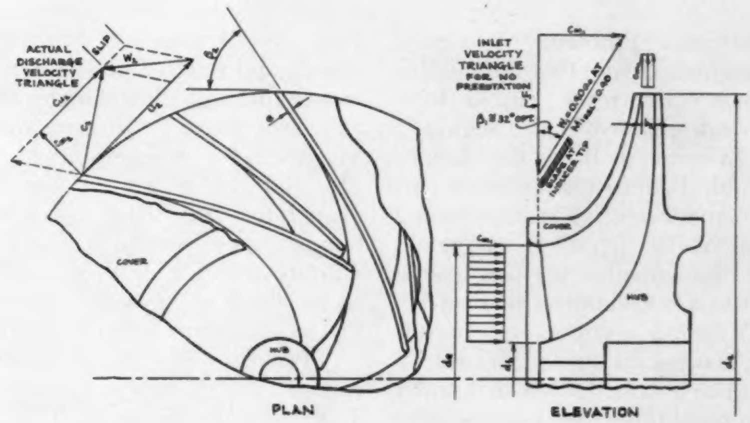

Fig. 8 Significant impeller dimensions and inlet and discharge velocity triangles

physical properties of each refrigerant, and the assumed standard cycles shown in Figs. 5 and 6, result in the compressor requirements listed in Table II for Refrigerants-11, 12, 113 and 114. Although these analyses can be undertaken for any load, we have chosen a typical load at 300 ton of refrigeration. Note the appreciable variations in the head, weight flow, and suction volume requirements for the four refrigerants at the constant load and temperature "lift" requirements.

Before entering into the determination of compressor dimensions, speeds and relative power requirements, we must make some further general assumptions with respect to the compressors as follows:

Maximum Permissible Inlet Relative Mach Number – (at the inducer or impeller eye tip) usually set at 0.90 for the highest flow compressor in any given "family" of machines, although a more conservative design would limit this value to 0.80 or less.

Inlet Prerotation – The information of Fig. 3 is intended for stages having no inlet prerotation, and must be adjusted when either with rotation or against rotation velocity components are induced at the impeller inlet (usually with the aid of fixed or variable inlet guide vanes, although prerotation can be "self induced" by improper inlet designs).

Optimum Inlet Flow Triangle (at the inducer tip) – This can be shown to vary principally with the assumed Inlet Relative Mach number and the Specific Heat Ratio of the vapor. Over a wide range of these variables, however, the optimum inlet angle varies only a small amount from an average value of 32° with the tangent.

Inlet Free Area Factor—In order to size the impeller eye or inducer tip diameter, we must assume some average restrictive effect of the impeller hub and blades. Also, there is a certain amount of expansion of the vapor as it flows from the evaporator to the impeller inlet. A factor of 82.5% free area may be assumed as representative of a typical practical design. This factor, of course, must be carefully evaluated for each specific final stage design.

Allowable Deceleration Rate of the Radial or Meridional Velocities from Impeller Inlet to Outlet—This is commonly specified at $K = c_{m_2}/c_{m_1}$ = no less than 1.00 for radial bladed impellers, ($\beta_2 = 90°$). For impellers with backward swept discharge blade angles, several investigators agree that a certain amount of meridional deceleration is permissible (for the same safe margin against flow separation in the impeller). Fig. 7 includes some representative values for allowable deceleration rates for backward swept impeller discharge blade angles, as a function of the angle β_2.

Taking these considerations or limitations into account, we may calculate the necessary stage dimensions, speeds, and relative power requirements for all of the common refrigerants under the conditions given in Table II. Table III shows the results for Single Stage Radial Bladed Units, while Table IV presents comparable figures for Two Stage Radial Bladed Units (with economizers). Obviously, the most compact and most efficient compressor in each case results with Refrigerant-12. The largest and least efficient compressor occurs with the choice of Refrigerant-113. Refrigerants-11 and 114 are nearly the same, and intermediate in these respects. When consideration is taken of the overall cycle performance, as represented by the Relative Gas Hp criterion, it can be seen that Refrigerant-11 should be expected to produce a somewhat better machine, followed by Refrigerants-12, 114 and 113, respectively. For machines operating in economizing cycles under the conditions assumed, it can be noted that the overall relative performance with the latter three refrigerants are all nearly identical (about 239 Relative Gas Hp).

The impeller tip flow coefficients for the single stage units listed in Table III all fall within the envelope of current experience shown in Fig. 3, however, the corresponding flow coefficients for the two stage units shown in Table IV are all beyond the limits of current experience, and may be expected to present some critical design and

TABLE III

APPROXIMATE DIMENSIONS, SPEEDS AND POWER REQUIREMENTS FOR 300 TON SINGLE STAGE (RADIAL BLADED) HIGH FLOW UNITS WITH REFRIGERANTS-11, 12, 113 AND 114

REFRIGERANT	11	12	113	114
Peripheral Speed, u_2, f_{ps}	640	566	579	530
Rotational Mach Number, u_2/a_o	1.450	1.250	1.554	1.388
Imp. Tip Flow Coefficient, ϕ_2	.323	.369	.302	.340
Speed, rpm	7610	18,170	3730	8190
Eye Diam, in., d_e	10.20	4.40	17.50	8.20
Imp. Tip Diam, in., d_2	19.28	7.15	35.50	14.83
d_e/d_2	.529	.615	.493	.553
Relative Polytropic Efficiency	.926	.979	.893	.949
Relative Gas Horsepower	256	261	272.5	269.5

TABLE IV

APPROXIMATE DIMENSIONS, SPEEDS AND POWER REQUIREMENTS FOR 300 TON TWO STAGE (RADIAL BLADED) HIGH FLOW UNITS, USING ECONOMIZERS, WITH REFRIGERANTS-11, 12, 113 AND 114

REFRIGERANT	11	12	113	114
Peripheral Speed, u_2, f_{ps}	460	410	419	385
Rotational Mach Number, u_2/a_o	1.040	0.905	1.124	1.010
Imp. Tip Flow Coefficient, ϕ_2	.450	.510	.418	.466
Speed, rpm	8000	19,570	3960	8910
Eye Diam, in., d_e	9.70	4.09	16.50	7.53
Imp. Tip Diam, in., d_2	13.19	4.80	24.20	9.93
d_e/d_2	.735	.851	.681	.758
Relative Polytropic Efficiency	.968	.992	.951	.979
Relative Gas Horsepower	233	238.5	239	238.7

performance problems. It should be mentioned here that the relative efficiency trends of Fig. 3 have been extrapolated in this region.

In order to bring the designs of Table IV into the region of current experience, while maintaining radial bladed impellers, we could limit the impeller tip flow coefficient to a maximum value of $\phi_2 = 0.375$ for all refrigerants. In this case, the results shown in Table V would be obtained. Note that under these conditions, the relative inlet Mach Numbers have been materially reduced, the rotational speeds have been decreased (with corresponding diametral increases), and the stages have moved into a region of performance where the efficiencies may be expected to improve. Again, on a Relative Gas Hp basis, Refrigerant-11 is still best, followed by Refrigerants-12, 114 and 113, respectively.

We may also bring the designs of Table IV into the region of current experience without changing the inlet relative Mach Numbers (except for Refrigerant-12) by considering backward swept impeller blading, at the same time allowing some nominal amounts of meridional deceleration as suggested by Fig. 7. In this case, the values shown in Table VI are typical of the result. These stages are somewhat larger than those in Table IV, but somewhat smaller than those in Table V. The relative efficiencies and corresponding power values are, however, comparable to those in Table IV, except for Refrigerant-12, where the inlet relative Mach Number was further limited (to 0.823), only to produce a design within the current range of experience.

The figures in Tables III and VI would be representative of the final overall impeller dimensions, speeds and relative power requirements for high flow Single Stage and Two Stage (with economizer) Machines for Refrigerants-11, 12, 113, and 114, at 300 ton and 34 F evaporator and 104 F condensing temperatures. Similar analyses are entirely possible, with the aid of these charts, at any other load and lift requirements for any vapor.

Professor Soo[9] has presented a similar analysis, where he assumed a pressure coefficient of 0.70, and an overall adiabatic compressor efficiency of 78% (the same for all refrigerants). His results were presented for Two Stage Units, with economizers, at 0.90 inducer tip Mach Numbers and operating between 38 F and 105 F. Without so stating, his assumption of 0.70 pressure coefficient obviously limits the application of his results to radial bladed stages only. Also, as with our Table IV, the impeller dimensions and speeds given in Professor Soo's Figs. 1, 2 and 3 are considerably beyond the envelope of current experience. So far, these assumptions do not violate the basic intents of his paper; i.e., the presentation of a sequence of maximum design capacities for various refrigerants at various speeds and Inducer Tip Mach Numbers. In assuming a constant compressor efficiency for all of these various possibilities, however, Professor Soo has failed to complete the story and indicate the predominant effects of Flow Coefficient, Rotational Mach Number and Reynolds Number on the relative efficiency levels or relative power requirements of the various vapors. This could be done now with the aid of Fig. 3, by projecting the polytropic equivalent of the 78% adiabatic efficiency, assumed by Professor Soo, into the framework of this chart at his Flow Coefficient and Rotational Mach Number for (say) Refrigerant-12. The corresponding Relative Polytropic Efficiency of Fig. 3 consequently takes on this assumed absolute value, and all other points on the chart are then defined in terms of this value. In like manner, absolute values can be assigned within the framework of Fig. 4. When this is accounted for, along with the other conditions assumed by Professor Soo, the comparable adiabatic efficiencies which may be expected for the four most common refrigerants would be similar to the following:

Refrigerant-11 – 76.6%
Refrigerant-12 – 78.0% (the assigned value)
Refrigerant-113 – 75.4%
Refrigerant-114 – 77.6%

In addition to allowing an analysis to be made for any impeller tip blade angle, the material presented in this paper will allow evaluations of designs which do not violate the current practical range of experience, and at the same time will distinguish between vapors as far as obtainable compressor efficiencies are concerned.

SUMMARY

It is felt that this paper presents that significant material necessary to specify basic impeller design dimensions and speeds of any centrifugal impeller, for use with any refrigerant vapor. At the same time, it is also possible to take advantage of those features which will govern the attainment of the best peak stage efficiencies. Obviously, as far as the compressor is concerned, the best efficiency will produce the least gas horsepower per ton of refrigerating effect. It is assumed that all major properties

TABLE V

APPROXIMATE DIMENSIONS, SPEEDS AND POWER REQUIREMENTS FOR 300 TON TWO STAGE (RADIAL BLADED) "CONSERVATIVE" FLOW UNITS, USING ECONOMIZERS, WITH REFRIGERANTS-11, 12, 113 AND 114. (DESIGNS AT $\phi_2 = 0.375 =$ CONSTANT)

REFRIGERANT	11	12	113	114
Peripheral Speed, u_2, f_{ps}	460	410	419	386
Rotational Mach Number, u_2/a_o	1.040	.905	1.124	1.010
M_1, relative	.737	.642	.795	.717
Eye Diam, in., d_e	10.60	4.77	17.45	8.40
Speed, rpm	5960	11,870	3300	6330
Imp. Tip Diam, in., d_2	17.72	7.92	29.05	13.96
d_e/d_2	.598	.602	.600	.601
Relative Polytropic Efficiency	.980	1.018	.957	.995
Relative Gas Hp	230	233	237.5	234.5

TABLE VI

APPROXIMATE DIMENSIONS, SPEEDS AND POWER REQUIREMENTS FOR 300 TON TWO STAGE HIGH FLOW UNITS (WITH BACKWARD SWEPT IMPELLER BLADING), USING ECONOMIZERS, WITH REFRIGERANTS-11, 12, 113 AND 114. (DESIGNS JUST AT LIMITS OF CURRENT EXPERIENCE)

REFRIGERANT	11	12	113	114
Peripheral Speed, u_2, f_{ps}	508	486	435	428
Rotational Mach Number, u_2/a_o	1.149	1.073	1.169	1.120
Imp. Tip Flow Coefficient, ϕ_2	.390	.345	.395	.395
Imp. Tip Blade Angle, β_2	65°	50°	80°	65°
Speed, rpm	8000	17,180	3960	8910
M_1, Relative	0.90	0.823	0.90	0.90
Eye Diam, in., d_e	9.70	4.21	16.50	7.53
Imp. Tip Diam, in., d_2	14.57	6.49	25.17	11.00
d_e/d_2	.666	.649	.655	.685
Relative Polytropic Efficiency	.966	1.002	.950	.979
Relative Gas HP	233.5	236	239.5	238.5

which tend to influence the peak efficiency of any stage, with any vapor, are adequately described and represented in Figs. 3 and 4 of this paper. The Relative Polytropic Efficiencies shown on these plots may be converted to representative absolute values by assuming a practical nominal value for some point within the field of Fig. 3 (as mentioned in connection with our discussion of the paper presented by Professor Soo).

With respect to the 300 ton comparisons calculated for Refrigerants-11, 12, 113, and 114, and presented in Tables III through VI of the paper, we may summarize the results as follows:

The most compact and most efficient compressor results with the use of Refrigerant-12.

In consideration of the overall machine performance, Refrigerant-11, with a somewhat poorer compressor efficiency, realizes the best Gas Hp per ton of refrigeration.

The largest and least efficient compressor is obtained with Refrigerant-113. Also, this refrigerant is the poorest on a Gas Hp per ton basis.

In the economizing cycle chosen for these comparisons, Refrigerant-11 still realizes the best power per ton, with savings of the order of 9 to 10% over the single stage cycle. Similar savings for the other refrigerants are found to be approximately as follows:

Refrigerant-12 – 9-11%
Refrigerant-113 – 14-15%
Refrigerant-114 – 13-15%

With regard to speeds at this load, Refrigerant-113 is most adaptable for direct motor drive, while the other refrigerants would require either speed increasing elements or high speed turbines, and possibly even both of these media in the case of Refrigerant-12. Obviously, it is this advantage of Refrigerant-113 which results in its use for this load range in direct drive hermetic compressors, in spite of the fact that its overall performance is not as good as that of the other refrigerants.

The charts presented in this paper should be most useful in correlating or evaluating the design considerations and the relative performance abilities of newer refrigerants as they come into existence for practical application in centrifugal refrigeration compressors.

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NOMENCLATURE

A =	flow area	ft²
a =	sonic velocity	ft/sec
b =	channel width	in.
c =	absolute velocity (or components, when used with subscripts)	ft/sec
C_p =	specific heat of vapor at constant pressure	Btu/lb/F
C_v =	specific heat of vapor at constant volume	Btu/lb/F
d =	diam	in.
e =	blade thickness	in.
Ghp =	gas or hydraulic horsepower	hp
g =	gravitational constant	32.17 ft/sec
h =	enthalpy	Btu/lb
H =	head	ft
K =	impeller meridional deceleration rate	dimensionless
k =	specific heat ratio, C_p/C_v	dimensionless
M =	Mach Number	dimensionless
N =	speed	rpm
Q =	volume flow	ft³/min
R_e =	Reynolds Number	dimensionless
u =	peripheral speed	ft/sec
v =	specific volume of vapor	ft³/lb
W =	relative velocity	ft/sec
w =	weight flow	lb/min
Z =	number of blades	——
β =	relative flow angle (or blade angle, when there are no inlet or discharge flow deviations), with the tangential direction, degrees	
η =	efficiency, expressed in %/100	dimensionless
μ =	head or pressure coefficient	dimensionless
μ =	absolute viscosity of vapor (= .000672 × centipoises)	ft-sec/lb
ρ =	density	lb/ft³
ϕ =	flow coefficient	dimensionless

(Continued on page 104)

Wall heat-flow measured

with large scale apparatus

K. R. SOLVASON

Built-up 8 x 8-ft wall sections can be tested to determine heat transmission coefficients under steady-state conditions; also, with a periodic variation in cold side temperature. Here is a definitive review of special apparatus developed for the purpose, together with a review of earlier and present design criteria.

Consisting of two 8 x 8 x 4-ft boxes, open on one side, and between which the test wall is placed, apparatus for the study and measurement of steady-state and periodic heat flow through 8-ft^2 wall sections has been developed by the Div of Building Research.

One box is maintained at the desired constant cold side temperature from –35 to +50 F for steady-state tests or varied according to some predetermined cycle for periodic heat flow tests. The other box is electrically heated to maintain a constant warm side temperature of from 65 to 75 F. The heat transmission coefficient for the wall specimen is calculated from the measured electrical input and temperatures.

In the design of the apparatus an attempt was made to overcome the limitations of the following methods for determining heat transmission coefficients: calculations using predetermined thermal conductivities or conductances of the various components and air films; the ASTM guarded hot box method; and the use of heat flow meters.

K. R. Solvason is Associate Research Officer, Div of Building Research, National Research Council (Canada). This paper is a contribution from the Div of Building Research and is published with the approval of the Director. It was presented as "Large Scale Wall Heat-flow Measuring Apparatus" at the ASHRAE annual meeting, Lake Placid, N. Y., June 22-24, 1959.

PRESENT METHODS

Many wall combinations are difficult to assess by means of calculation. One reason is that the only reliable thermal conductivities available for most materials are those obtained by hot plate tests on dry materials and these may be considerably different for actual operating conditions. Also, surface conductances are not necessarily constant over the whole of a wall surface and the average conductance will depend on the temperature difference between the wall surface and air, and that between the wall surface and its surroundings. A third reason is that most wall combinations contain heat flow paths of differing conductance, the effect of which is difficult to assess without resorting to a rigorous mathematical treatment or to analogue techniques, because of lateral heat flow in some of the components.

The ASTM guarded hot box[1] method consists essentially of placing an electrically heated metering box over the center portion of the warm side of the test wall. Surrounding the metering box is a larger guard box. The temperatures in the two boxes are adjusted so that there is no differential (and hence no heat flow) across the walls of the metering box. The conductance is calculated from the electrical input to the metering box and the temperature difference across the wall.

The foremost disadvantage of this method is that the metering box interferes with the convection over the test wall, so that forced convection must be resorted to and this may give film coefficients different from those occurring in practice. It is difficult to produce equal coefficients for the metering area and the guard area, so that lateral heat transfer may occur from the measuring area to the guard area.

A second disadvantage is the fact that the metering box placed over the central portion of the test wall measures only the heat flow into that portion, but it has been shown by G. O. Handegord and N. B. Hutcheon[2] that this is not necessarily the average heat flow for the whole test wall, especially for walls containing vertical air spaces; blocking of air spaces in the test area may change the conductance substantially.

Another disadvantage to be considered is the fact that the radiation exchange is indefinite and it is difficult to produce the same effect on both the metering area and the guard area of the test wall. Differences in the radiant exchange from the inner and outer surfaces

of the metering box may require that different air temperatures be provided in the test box and the guard box in order to maintain zero heat flow across the test box walls and this may lead to lateral heat transfer in the test wall.

Finally, in many cases the metering box will not cover a representative complete module of the test wall.

There are many reasons why the use of heat flow meters of the multiple differential thermocouple type is considered inadequate for the measurement of heat transmission coefficients. The heat flow must be integrated over the whole wall surface and an accurate weighted average is difficult to obtain on walls where large variations in heat flow occur, and the meter, if placed on the wall surface, probably interferes with the convective transfer.

Another reason for the inadequacy of heat-flow meters of this type arises from the fact that the thermal resistance of the meter itself reduces the heat flow in the area covered by the meter. Although corrections for this effect can be calculated, in some cases, they are probably not reliable when applied to walls that have surfaces of high conductivity material covering low conductivity material, since heat transfer in the plane of the wall surface permits part of the heat to bypass the meter.

Again, good contact between the meter and wall surface is often difficult to achieve and air spaces between the meter and wall surface will increase the thermal resistance through the meter-wall combination; heat flows in the plane of the meter to the points of contact are possible so that, if the thermopile element is localized in (say) the center of the meter, the heat flow might bypass the element or the heat flow through it might be exaggerated, depending on where contact occurs.

CRITERIA FOR DESIGN

With consideration for the inadequacies and disadvantages of the old methods the heat flow apparatus was constructed to meet the following requirements. First, it was designed to meter the heat flow into the whole of representative sections of building walls, and

Fig. 1 Apparatus consists essentially of a cold box and guarded hot box

to expose the test wall to controlled temperature surroundings as well as to air at controlled temperature. Also, it was built to operate without forced air circulation over the warm side of the test wall in order to produce warm-side film coefficients approaching those occurring in practice. Finally, it operates with the cold-side temperature either constant or varying according to some predetermined cycle.

Apparatus consists essentially of a cold box and a guarded hot box combination, with an 8-ft square measuring area. This size was chosen in order to accommodate walls with air spaces 8 ft high and with modules up to 4 ft wide. It was decided to limit the depth of the box to 4 ft for practical reasons. This was considered large enough to produce natural convection effects, approaching those occurring in practice.

The warm box was constructed with a liner of aluminum panels (Fig. 3) similar to those used to line the ASHAE environmental rooms. These panels, which were developed for use in radiant heating and cooling applications, consist of aluminum extrusions with holders for copper tubing, through which liquid is circulated. The inside of the box was painted to increase the emissivity of the panels to about 0.9. A second set of panels was installed outside of the liner and separated from it by thermal insulation. When the outer panel is maintained at the same temperature as the inner panel, it forms a guard to prevent any heat transfer across the walls of the box. Insulation (2-in. mineral wool batts) and a plywood covering were applied to the outside of the guard panel.

A water reservoir, pump, and pump motor for circulating water through the tubes of the inner panel are located inside the box. The pump is belt driven by a series-wound dc motor. The voltage-current characteristics of this type of motor permit the energy input to be varied from 25 watt (85 Btu per hr) to 500 watt (1700 Btu per hr) by regulating the voltage from 15 to 90 volt. Direct-current power was considered easier to control and to measure accurately than ac. The electrical input to the pump motor is normally sufficient to provide all of the wall transmission; thus the energy imparted to the water by the pump serves to heat the panel and the energy loss from the motor and drive serves to heat the air directly. Air heaters and water heaters are also provided so that the ratio of air heat to panel heat may be varied if desired.

Twelve parallel water circuits (each circuit supplying 4 by 4 ft of panel) are used to supply the panels in order to permit the use of a large quantity of water without excessive pump head and pump power. At the maximum input of 1700 Btu per hr, about 20 gpm of water are circulated. Since only about 60% or less of the total power input will be supplied to the water, the drop in temperature passing through the panel is only about 0.085 F. The ratio of water quantity to input increases as the total input decreases, so that the temperature drop through the panel is always less than 0.085 F. It can also be shown that the panel to water temperature difference and also the temperature variation over the panel surface is quite small, not more than 0.1 F.

The water reservoir is incorporated into a shield in order to prevent the test wall or the box walls from "seeing" the pump motor, which will operate at some temperature higher than the panel.

A second reservoir and pump located outside of the warm box supplies water to the outer guard panel. The reservoir contains a cooling coil and an electric heater

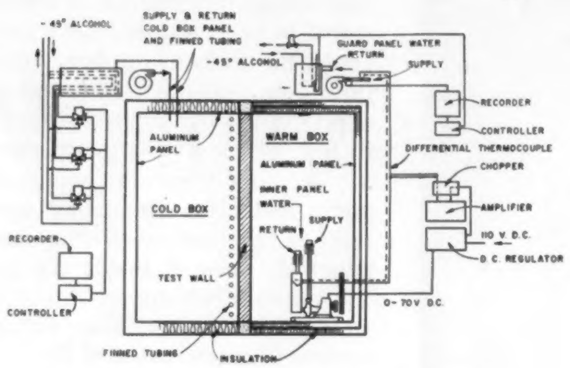

Fig. 2 General arrangement of components

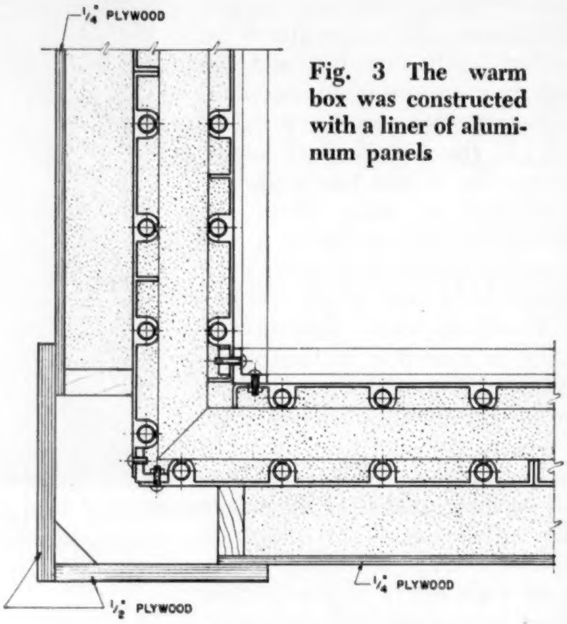

Fig. 3 The warm box was constructed with a liner of aluminum panels

to control the water temperature.

The cold box is simply a well-insulated box with an aluminum panel liner and a sheet metal exterior vapor barrier covering. The panel liner serves a three-fold purpose: to provide controlled temperature surroundings to which the test wall is exposed; to provide some convective heat exchange area; and to intercept heat gains to the box itself and thus reduce the heat exchange area required inside the box. Additional heat exchange area (312 sq ft surface area of finned tubing) is provided to bring the air temperature as close as possible to the panel temperature. Liquid (1/3 water, 1/3 ethylene glycol and 1/3 methyl alcohol) at the desired temperature is circulated through the tubes of the panel liner and the finned tubing. This liquid is cooled in a shell and tube heat exchanger by a low temperature liquid (−45 F methyl alcohol) from a central low temperature chiller serving other laboratories in the building. The heat exchanger contains eight 3/8-in. outside diam copper coils 20 ft long. The control valves are arranged so that the cold liquid can be circulated through 2, 4, 6, or all 8 coils depending on the temperature difference desired.

INSTRUMENTATION

The control and measuring instruments for the apparatus were selected in order to limit any specific error to as low a value as practicable, or to a maximum of one percent at the minimum heat flow and temperature difference. A heat flow of 200 Btu per hr and an overall temperature difference of 30 F were selected as a design basis. Errors in the determination of an over-all heat transmission coefficient may arise from errors in measurement, in control of temperatures and power input, and from heat leakage.

Temperature measurements are made with 30-gauge copper-constantan thermocouples and an electronic self-balancing temperature indicator. Checks of the indicator and thermocouple wire indicate that an accuracy of ± 0.05 F can be obtained. The temperature difference measurements are thus accurate to about ± 0.1 F. The percentage error is then about ± 0.33 percent at an overall difference of 30 F.

Warm box control is required to adjust the input voltage to the warm box to a stable value such that the heat input equals the wall transmission within 2 Btu per hr or one percent, and to maintain the guard panel at a temperature such as to prevent a heat transfer of more than 2 Btu per hr across the box walls. The panel liner of the warm box and the water have a large thermal capacity, about 330 Btu per F. The maximum permissible temperature change is, therefore, only 0.006 F per hr in order to limit heat storage to 2 Btu per hr.

The insulation between the inner and guard panels provides a calculated conductance of approximately 0.125 Btu per (hr) (sq ft) (F temperature difference) or a total heat flow of 24 Btu per (hr) (F temperature difference). Thus for precision of one percent, the average temperature difference must be controlled to about ± 0.08 F. A moderate ripple in the outer panel temperature can be tolerated, provided the average temperature is within the ± 0.08 F, since the insulation between the inner and outer panel will provide sufficient damping to prevent any appreciable heat flow at the inner panel.

The control system is shown in block form in Fig. 1. A thermocouple actuated electronic recorder with a three action (proportional + reset + rate) controller is used to control the guard panel temperature. This controller regulates a solenoid cooling valve and electric heater with manual switches, so that heating and cooling can be supplied continuously on, continuously off, or pulsed on-off by the controller. The dc input to the inner panel pump is supplied by a dc power supply[3], the output voltage of which is regulated from 0 to 70 volt by the amplified unbalance from a differential thermocouple to maintain equal inner and guard panel temperatures.

The differential thermocouple signal is first converted to a square wave by a mechanical chopper, amplified by a high gain low level dc amplifier and rectified by the chopper. The higher voltage dc signal is then used to regulate the

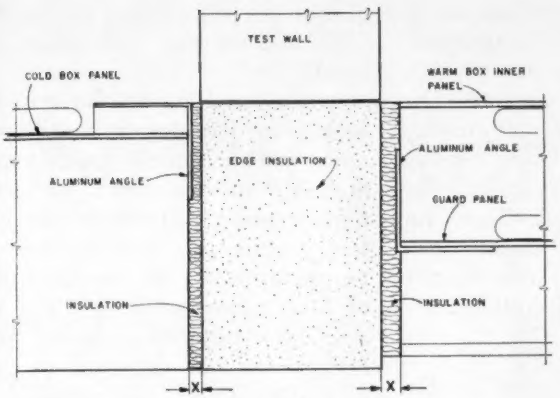

Fig. 4 Overcoming heat edge leakage

output voltage of the power supply.

This system has been used with only moderate success. Slight fluctuations in the guard panel temperature produce large fluctuations in the inner panel power. The effect of rapid fluctuations has been eliminated successfully by applying thermal damping to the guard panel thermocouple junction. Lower frequency temperature swings produce a similar swing in inside power. This effect, which is not serious for steady-state tests but unsuitable for periodic tests, could be eliminated by converting the present thermocouple recorder-controller to a shorter span and more accurate resistance-element-type.

Some additional difficulty has, however, been experienced with the chopper-amplifier system. The chopper point adjustment is difficult to maintain over long periods and this, together with stray inductive pickup, produces fluctuations in the power and errors difficult to assess. The present power supply, in addition, is not of sufficient capacity for the maximum inputs required. The control and power system is therefore being replaced with equipment which has recently become available.

A magnetic amplifier of 500-watt maximum capacity is being procured for a power supply. The magnetic amplifier regulated by the present recorder (converted to a resistance type) and a current adjusting controller will be used to control the inner panel temperature. The recorder sensitivity will be approximately 0.001 F and the combination is expected to control to 0.002 to 0.003 F. The error due to the capacity effect of the box is thus expected to be no more than 0.5 percent at the minimum heat flow.

The guard panel temperature will be controlled by an on-off electronic temperature controller which has a sensitivity of 0.001 F and is expected to control temperatures to well within ± 0.03 to 0.04 F of the inner panel. The unbalance errror is then expected not to exceed about 0.5 per cent.

The input voltage and current are measured by a millivolt recorder connected across appropriate resistors. The maximum recorder error, with calibration, can be limited to less than ± 0.25 percent of full scale so that by sizing the resistors to give larger than half-scale deflections this error should always be less than 0.5 percent. The error in voltage times current is less than 1.0 percent.

Cold box temperature control offered no special problems. Here a thermocouple-actuated electronic recorder with a three-action temperature controller is used to control one, two, or three solenoid valves which regulate the flow of cold liquid through the heat exchanger. The number of valves used is selected manually, depending on the temperature, to make 1/4, 1/2, 3/4 or all of the heat exchanger area effective. With this combination, the cold-side temperature can be held constant to about ± 0.1 F or about 0.33 percent of the temperature difference.

The controller is now equipped so that an approximate sine wave variation in temperature can be produced with amplitudes of 5 to 15 F and periods of 6, 12 or 24 hr. A program controller may be added later with which any desired wave form can be produced.

If all the measurement and control errors were additive, the total error is expected to be less than 2.0 percent. Operation to date indicates that heat transmission coefficient values can be reproduced within 1.0 percent. Tests to assess measurement errors experimentally will be conducted in the near future.

EDGE HEAT LEAKAGE

Heat leakage at the edge of the test wall is another source of error. The apparatus is located in a laboratory where the temperature is maintained close to the usual warm-box operating temperature. Heat leakage will, therefore, tend to occur through the edge of the test wall into the cold box. The edges of the test wall will usually be surrounded with low conductivity material (k = 0.25) slightly thicker than the test wall and some 5-1/2 in. wide (the width of the box edges) as shown in Fig. 4. Aluminum angles attached to the cold box panel and extending 2 in. over the box edge are installed to extend the low temperature plane outside of the test wall area. A similar angle was attached to the guard box panel so that the heat loss for this portion will be supplied by the guard panel rather than the inner panel.

The additional thickness of insulation (x in Fig. 4) should be such that its thermal resistance approximately equals the air-film resistance, so that the temperature in the plane of the wall surface will equal wall surface temperature. In many cases, the arrangement of the edges of the test walls may be such that good edge guarding is quite difficult. It may therefore be necessary to use test sections smaller than 8 ft square and use a filler of known thermal properties.

It would be quite difficult to calculate the error introduced by edge leakage, except perhaps by one of the relaxation or analogue methods. Tests conducted to date indicate no significant temperature gradients on the test wall surfaces near the edges, so that this error can be assumed to be rather small. Further tests to assess edge effects

will be carried out in the near future.

IN OPERATION

The total heat transfer at the warm-side surface of the test wall will be the sum of the net radiant exchange from the box panels to the test wall plus the convective exchange from the air to the test wall.[4] This can be expressed as:

$$q_t = q_r + q_c = F_c F_e \times 0.173 \left[\left(\frac{T_p}{100}\right)^4 - \left(\frac{T_w}{100}\right)^4 \right] + X(t_a - t_w)^y \quad (1)$$

The radiant and convective components are usually accounted for by coefficients h_r and h_c respectively, such that

$$q_t = h_r\ (t_p - t_w) + h_c\ (t_a - t_w) \quad (2)$$

The usual surface conductance coefficient, f_i, is the sum of h_r and h_c, hence,

$$q_t = f_i\ (t_i - t_w) \quad (3)$$

If the average air temperature (t_a) in the apparatus is not equal to the panel temperature (t_p), an equivalent temperature t_i, can be calculated from Equations (1), (2), and (3) which would produce the same heat flow as the actual t_p and t_a.

$$\text{Thus, } t_i = \frac{h_r t_p + h_c t_a}{h_r + h_c} \quad (4)$$

CONCLUSION

This apparatus is expected to provide an accurate and realistic means for obtaining steady-state heat flows or conductances through built-up wall sections, windows, and doors. The sample size used in the apparatus is considered large enough so that the results, especially for inside surface conductances, will be representative of much present-day construction.

The test wall is exposed to controlled temperature surroundings as well as to controlled temperature air. The convective and radiant components of the surface conductances can thus be evaluated.

The apparatus also provides a means for assessing the transient response of wall sections. In this respect the effect of moisture movement induced by cycling temperature can be investigated.

The apparatus can also be adapted to general calorimetry tests such as the lag characteristics and maximum air conditioning loads from appliances. Tests for water vapor and air transfer can also be carried out without any major additions or alterations.

ACKNOWLEDGEMENTS

The author is indebted to Dr. N. B. Hutcheon, Assistant Director of the Div of Building Research, and to A. G. Wilson, Head of the Building Services Sect, for their many helpful suggestions in the design of the apparatus and in the preparation of this paper, and to H. L. Egan, laboratory assistant, who performed most of the construction work.

NOMENCLATURE

- q_t = total heat flow rate, Btu per (ft²) (hr)
- q_r = net radiant exchange, Btu per (ft²) (hr)
- q_c = convective exchange, Btu per (ft²) (hr)
- F_c = the form factor for radiant exchange between the box panels and the test wall
- F_e = emissivity factor
- T_p = box panel temperature, Fahrenheit absolute
- T_w = average test wall temperature, Fahrenheit absolute
- t_a = average air temperature, F
- t_p = box panel temperature, F
- t_w = average wall surface temperature, F
- t_i = temperature, F, equivalent to t_a and t_w
- X = constant
- y = constant
- h_r = surface coefficient for radiant heat transfer Btu per (hr) (ft²) (F)
- h_c = surface coefficient for convective heat transfer Btu per (hr) (ft²) (F)
- f_i = combined surface coefficient Btu per (hr) (ft²) (F)

The form factor F_c for the test wall and box can be shown to be equal to 1.0, and the emissivity factor F_e to equal $e_p e_w$ where e_p and e_w are the emissivities of the box walls and the test wall surface respectively.

If e_p and e_w are known, the factors for convective heat flow x and y in Equation (1) can be evaluated by measurements of panel temperature (t_p), average test wall temperature (t_w), and average air temperature (t_a) at several values of heat transmission (q_t). For the range of conditions expected, these factors are not expected to vary so that they need be evaluated for one wall only. In any later tests, if the average wall surface temperature can be measured, it will be possible to calculate the wall surface emissivity. For those walls where surface temperature variations make the assessment of the average difficult, it will be possible to calculate the average temperature if the emissivity e_w is known. The cold-side heat exchange can be treated similarly.

Thus the surface conductance can be accurately established, the average surface temperatures defined, and the conductance of the wall itself (without surface conductances) can be calculated. Over-all conductances for any other surface conductances can then be calculated.

The warm side surface conductance (f_i) resulting in the apparatus is expected to be quite close to the still air conductances that result in practice, if a wall is exposed to air and surroundings at nearly equal temperatures. This coefficient is not a constant but rather a function of the temperature level, the temperature difference, and the wall emissivity.

The cold-side surface conductance coefficients in the apparatus are expected to be much lower than will be desired in most cases. Forced circulation either by fans or induced by jets will be provided to increase the conductance as required.

Recent gifts to the John R. Allen Memorial Library, maintained by ASHRAE at the Society's Research Library, 7218 Euclid Ave., Cleveland 3, Ohio, include:

Proceedings of the Institution of Mechanical Engineers, Vol. 171, 1957, London.

Proceedings of the Institution of Mechanical Engineers—Automobile Div., 1955-56, London.

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Proposed Standard to measure

Sound from equipment

Sound conditioning – The purpose of comfort air conditioning is to provide a more favorable environment for people. More specifically, it deals with the conditioning of the air surrounding people. The definitions of air conditioning include the control of the temperature, humidity, cleanliness, mass motion and quality (with respect to contaminants).

But, they omit any mention of one important characteristic of the air: the vibratory motion known as sound. Objectionable sound can come from a wide variety of sources, including traffic noise, typing, talking, mechanical equipment and even from air conditioning and refrigeration. Contributing to the sound can be such factors as poor building construction, improper sound treatment of offices, flimsy partitions, cracks under doors and many other factors which permit undesired sounds to reach the ears of building occupants.

Absence of sound can be just as objectionable as too much or the wrong kind of sound. Sounds of good quality can be used to mask out objectionable noise and can protect private conversations and prevent intrusion of traffic and typing noises. Thus, we are beginning to learn that what is needed is not sound removal, but sound control. In fact, "sound conditioning" may be considered as the new tool for improved human comfort.

Air conditioning which includes elements providing for heating, ventilating and cooling can and should be an important part of sound conditioning. But, frequently the air conditioning is charged with the overall responsibility for the sound conditioning. Other factors, including building construction, other mechanical equipment and the installation of

C. M. ASHLEY
Fellow ASHRAE

the air conditioning equipment must bear a major share of the responsibility. An important reason for knowing the sound output of the air conditioning equipment itself is so that a proper division of responsibility can be made between sound which is made by the air conditioning equipment and sound coming from other sources.

The sound standards program – The sound standards program is the result of a growing recognition of the fact that we cannot hope to get proper sound control with respect to air conditioning and refrigeration without having adequate means of objective measurement, any more than we could get adequate temperature and humidity control without being able to measure the cooling and heating power required to effect them. The fact that we do not have universally adequate control over air conditioning sound is amply attested by the results of user surveys such as those of duPont[1] and Architectural Forum[2] and by our own personal experiences with noisy equipment.

As an important step in overcoming this situation, ASHAE in 1955 activated a sound study program in its research laboratory. The initial project was to determine the techniques of measurement of sound power output of fans.[3] The program is being continued at the present time with a study of sound generation and attenuation in duct systems. In 1957, ASRE took the leadership in setting up a standards committee to prepare a standard for sound testing of equipment and invited ASHAE to participate in the standard on a joint basis. A task committee was appointed in July, 1957, and has prepared several drafts of a primary test standard.

In accordance with recognized practice, the Society standards committee writes a standard test procedure, but does not ordinarily set up equipment ratings or other recommended practices. In order to make the standards activity effective, additional steps are required of trade associations or other organizations. Among the steps which are pertinent to a sound standard are:

1. A recommended basis of rating for different types of equipment.
2. Interpretation of the ratings in terms which are meaningful to the user of the equipment.
3. Recommended sound criteria for different space uses.

In undertaking its assignment, ASHRAE was fully conscious that there was a need for much more widely diffused knowledge and understanding of the technical aspects of sound and that there must be a step-by-step development of the procedures rather than attempting to reach the ultimate immediately.

However, it was also recognized that if the standard or standards were to be of value, they must meet at least two criteria. First, the performance of equipment should be measurable in terms which would be significant regardless of the environment. Second, the results must be capable of reasonable correlation with the

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C. M. Ashley, Past President of ASRE, is Chief Staff Engineer, Carrier Corporation.

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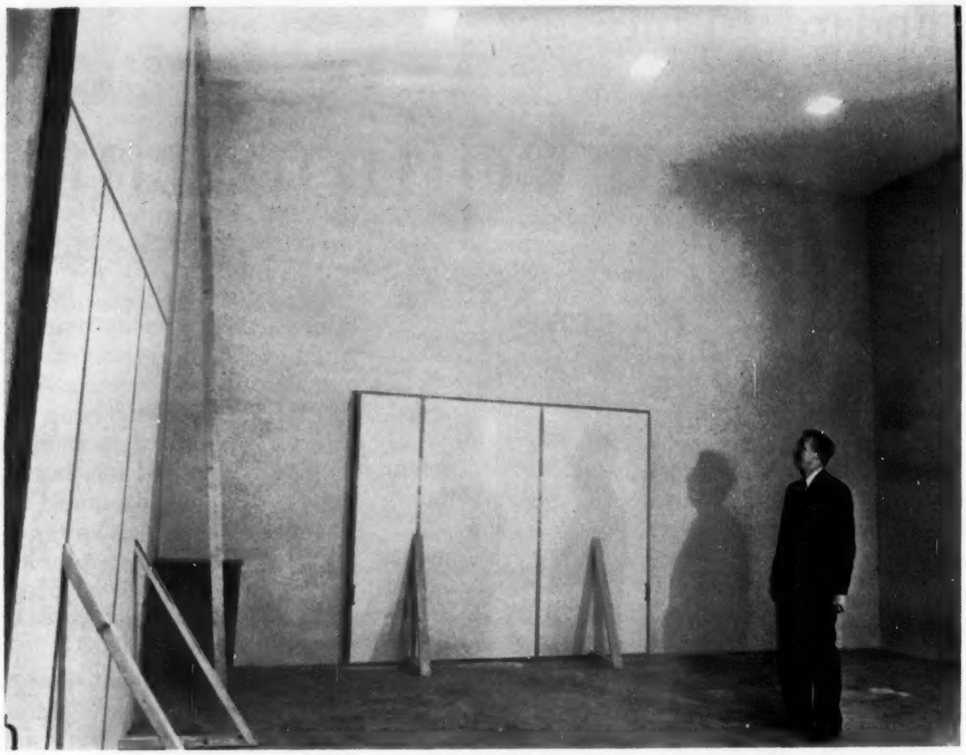
[1] Superscript numbers refer to references at the end of the paper.



Author Ashley examines a representative 10,000 cu ft hard sound room suitable for a reverberant room for testing of air conditioning and refrigerating equipment

subjective impressions of observers in spaces where the equipment is to be used.

FEATURES OF THE STANDARD

Sound power — In order to satisfy the first of these requirements, it was agreed that we must state our measurement of sound in terms of the sound power output of the equipment rather than the sound pressure level[4]* which has been customarily used in the past and which is the basis of most sound measurements in rooms. This is necessary because sound pressure level is a function not only of the sound output of the equipment, but also of the sound characteristics of the space in which the equipment is located. Using sound pressure level as a basis for rating equipment sound would be like using room temperature as the basis of rating the capacity of a cooling unit. Neither one deals with the performance of the equipment itself, but only with the results in a specific environment.

Because the total variation in the sound power output of different types of equipment is great and because the ear responds to sound on an approximately logarithmic rather than a linear scale, the power output of the equipment is expressed in logarithmic terms rather than in arithmetical terms as sound power level.* If the sound power output of the equipment and the acoustical characteristics of the room are known, then it is possible to make a reasonably accurate prediction of the sound pressure level to be expected as a result of the equipment sound, just as it is possible to make a reasonably accurate prediction as to the room temperature which will be achieved with a given cooling unit under given ambient conditions.

The sound pressure level of a room is usually in the range of 12 to 20 db below the sound power level of equipment radiating sound directly to the room.

Sound quality discrimination — The reaction of a person to the equipment sound is the result not only of the sound pressure level, but also of the frequency, time and space characteristics of the sound. In general, the characteristics of a sound which the ear recognizes as good are:

1. "Broad band," that is, having a continuous and fairly uniform frequency spectrum without noticeable single frequency or pure tone components.
2. Without special character or pitch to make the listener conscious of the sound. This effect is produced by broad band noise having decreasing sound pressure level per octave for successively higher frequency bands.
3. Continuous with respect to time rather than intermittent or varying in level.
4. Well diffused rather than being directional or concentrated close to the observer.

Fortunately, most sounds from aerodynamic sources, such as fans, tend to conform to the description given above for a good sound. Thus, the major problem with such equipment is to be able to maintain a proper level and to eliminate sources of poor quality noise generation such as whistles due to air leaks, blade cut off noise, magnetic motor noise, etc.

However, the sound from refrigeration equipment tends to have single frequency components due to motor hum, gas pulsations, mechanically excited vibrations, etc. Thus, the noise from the mechanical parts of the equipment are

* For definitions of acoustical terminology as well as much valuable background information, see Chapter 25, Sound Control, in the 1959 Heating, Ventilating and Air Conditioning Guide.

likely to be far less desirable in quality than is the air noise.

Frequency bands—In order to provide a measure of quality, the committee is recommending the determination of the sound power level on an octave frequency band basis. When these values are converted to room sound pressure level, the effect of the sound in the room may be interpreted in terms of various criteria such as the Noise Criteria,[5] the Neighborhood Annoyance Index,[6] Speech Communication[7] or Loudness[8] since all of these criteria are based upon octave band room sound pressure levels.

It is recognized that the presentation of the data in octave bands is not necessarily a complete solution to the problem because the octave band readings may not fully reflect the annoying quality of discrete frequencies, beats or time or space variable sounds. Nevertheless, it is believed that it is a good beginning and that it is especially appropriate in view of the prevailing characteristics of sound from air conditioning equipment.

Quite possibly at some later date, after experience has been gained with the use of the proposed standard and after analytical test equipment is more generally available, it may be desirable to change the requirement to a smaller band width, such as one third octave.[9] Also, it may be found advisable to add a requirement for a narrow band analysis to show up the presence of discrete frequencies. However, it is not believed by your committee that either of these steps is appropriate to the present situation.

Test room requirements—The committee next turned its attention to the question of the type of environment in which the test should be run. From the practical point of view, there are a number of factors which need to be considered. First, the cost of the test facilities should be kept as low as practicable consistent with good accuracy in measuring the sound. Second, the method of testing chosen should be simple enough so as to be usable by a person having relatively little background in sound testing and also should be relatively economical of test time. Third, the test space should be as flexible as possible for the testing of different types and sizes of equipment and should permit as far as practicable the use of existing facilities. Finally, and most important, the test facility together with the test technique chosen should be such that the results obtained provide the true power level and give reproducible results between different laboratories.

Anechoic room—The use of an anechoic room was considered. In such a room it is possible by taking a series of readings on an envelope around the equipment to obtain the sound power level directly from the readings of sound pressure level by taking into account the area of envelope on which the measurements are made. However, in an anechoic room we are faced with the same sort of problem in measuring sound as we face when measuring air quantity with a Pitot tube in a duct: if we are to have good accuracy, we must take a number of readings. In the case of the envelope measurement, it is probable that at least twelve stations would be required. Thus, unless automatic traversing equipment is available, the measurement becomes laborious and complex. In addition, the cost of an anechoic facility is quite substantial and it is not readily adaptable for or from other uses.

Semi-reverberant room—The committee next considered a "semi-reverberant" room which has a fair amount of sound absorption, possibly comparable to that in a normally occupied room and where both direct and reflected sound is received by the microphone.[10] Such a facility has an advantage in comparison with an anechoic room of lower first cost and greater utility for other purposes, but it will still require the use of a multiplicity of stations to provide the desired information. It is recognized that either an anechoic room or a semi-reverberant room is necessary if the directivity characteristics of the sound are to be measured. However, the committee felt that at least initially it was not desirable to include this additional requirement.

Reverberant room—The committee then considered the use of a reverberant test room in which it is possible for the sound to be averaged by multiple reflections from the relatively hard surfaces of the room. In such a room it should be possible to obtain the required data by means of a single station rather than the multiple stations of the traverse in the anechoic and semi-reverberant rooms. The reverberant room is essentially a room of suitable size and shape constructed of relatively hard nonabsorbent surfaces. However, for most air conditioning and refrigerating equipment there will also be a need to provide unusually good isolation from outside noises so that they will not interfere with the measurement of the sound from the equipment. This usually involves the use of a room of basic masonry construction resiliently mounted and possibly enclosed in an outer room.

There are a number of problems in connection with the testing of equipment in a reverberant room. Unfortunately, the sound is not reflected and diffused uniformly all over the room.[11] Particularly at low frequencies, the room tends to act in the same manner as an organ pipe, the sound being reflected back and forth between opposite surfaces. Since the room is three dimensional, these reflections may occur on a one dimensional, two dimensional or three dimensional basis and the resulting modal distribution patterns become quite complex.

At low frequencies there is an appreciable frequency separation between successive room modes and in order to space these as evenly as possible, it is important that the proportions of the room be kept within prescribed limits. In order to increase the number of modes at low frequencies, it is desirable to have the room as large as practicable. Even at higher frequencies, a single frequency sound produces a standing wave pattern in the room. Thus, it is desirable to have a swinging microphone which moves back and forth so as to be able to average the sound over an appreciable path length.

Microphone location—The microphone should be a reasonable dis-

tance away from the room surfaces because there is an increase in pressure level adjacent to the room surfaces and a further increase next to the edges between two surfaces and a still further increase in the corners. In addition, the microphone must be located far enough away from the equipment so that it will be in the "reverberant field," that is: so that the amount of sound which is radiated directly from the equipment to the microphone will be negligibly small in relation to the amount received by the microphone from the multiple room reflections.

It is fortunate that the sound being measured is generally broad band in character since the effect is to provide an averaging of the sound which largely submerges the unusual characteristics of the room. However, it must be recognized that where single frequency components are prominent in the sound spectrum of the equipment, the measurement in a reverberant room is less than ideal. This is especially true of single frequency components in the lower frequency region below 150 cps. A highly reverberant room may react with the equipment either to increase or diminish the sound power of the single frequency component. The magnitude of these effects is decreased as the amount of sound absorption in the room is increased. Thus, it is desirable to use as much sound absorption in the room as practicable and still retain the reverberant character of the room.

In order to satisfy the requirements listed above, a minimum room volume of at least 6,000 cu ft is proposed together with room dimensions of approximately 25 ft long, 20 ft wide and 12 ft high. Larger rooms are preferred and where the equipment size is large, may be necessary. A suggested series of dimensions are given in the standard and are shown in Table I attached. The table also recommends optimum sound absorption characteristics for the room.

Diffusion of sound in reverberant room – While it is accepted that the basic reverberant room recommended in this discussion is adequate for most of the sounds which will be measured, it is also recognized that in looking toward the future some additional means of diffusion must be provided.[12] One method of accomplishing this objective is to increase greatly the size of the room while retaining good proportions and sound absorption characteristics.

Two other approaches are being considered which are applicable to the smaller sized rooms. The first of these, which has been used in a number of acoustical laboratories, is to provide a large revolving reflecting vane which continually changes the effective room geometry through a substantial range. While it is probable that satisfactory results can be achieved by this means, it requires the use of a vane of dimensions not much smaller than the room dimensions. Thus, there is little room for the apparatus to be tested particularly so if it is bulky. In addition, there are considerable mechanical and acoustical complications in providing a large rotating vane which will also not generate measurable noise.

A second approach to the problem is to provide bumps or panels to increase the number of effective reflecting surfaces in the room. If these panels are to be effective for the low frequency sound, their dimensions must be a substantial part of a wave length, probably of the order of at least 8 ft or 10 ft. In addition, there is some uncertainty as to how far out from the walls they must be placed in order to be effective. There is a need for further research on means of diffusion and it is hoped that before the standard is ready to be issued a better evaluation will have been possible.

Accuracy of readings – If equipment ratings are to have any real significance, they must have a reasonably high degree of absolute accuracy or at least they must be capable of being checked in a series of laboratories. Since the differences in competitive equipment are likely to be of the order of 5 db or less and since a 5 db variation in sound pressure level is readily observed by a listener, this large a plus or minus variation would not be acceptable. A probable variation of plus or minus 2 or 3 db would still be undesirable, but possibly acceptable.

The need for accuracy poses a serious technical problem since there are a number of possible sources of error, some of which can be of substantial magnitude. These include deviations in the

TABLE I
TEST ROOM CHARACTERISTICS

Volume ft³	Range of Dimensions, ft Length	Width	Height	Minimum Distance Microphone to Acoustic Center[1] of Source, ft	Maximum Equipment Perimeter, ft	Room Adsorbtion Constant R, ft²
6000*	28-25	18-20	12°	13	20	150-210ˣ
7500	25	20	15⁺	14	20	175-250ˣ
	32-28	21-24	15			
10000	27½	22	16½⁺	15	25	200-300ˣ
	25	21½	18			
	42-39	32-34	12°			
16000	36	28	16	20	33	300-450ˣ
	32	26	19⁺			
	51-47	41-45	12°			
25000	46-43½	34-36	16	25	50	400-600ˣ
	45-38	28-33	20			

* Minimum volume.
\+ Most desirable proportions.
° Minimum height.
[1] Acoustic center of source is the projection of the physical center line of the equipment to the nearest room surface or the intersection of the two nearest room surfaces when close to both.
x Room absorption constant R corresponds to a range of sound absorption coefficients of 0.07 to 0.10. The sound absorption coefficient and room constant should be a maximum in the first and second octave bands and should be less than the range indicated in the higher bands. This absorption characteristic can be obtained by the use of hard, relatively light panels well spaced away from the walls. The room absorption constant can be determined experimentally with a calibrated standard sound source and sound level meter according to Section 2.4 and equation (4)

$$L_w = L_p + 10 \log_{10} \frac{R}{4} - 0.5 \text{ db}. \qquad (4)$$

sound level meter and octave band analyzer used for the sound measurement, the variations of sound pressure level in the reverberant room, effects due to the frequency characteristics of the equipment, observational errors and a number of others. In the past it has been the practice to determine the sound absorptive characteristics of the measuring room and then determine the sound power level indirectly from these characteristics and a measurement of the sound pressure level. Such an indirect method involves all of the major sources of error listed above.

Standard sound source – In order to improve accuracy and eliminate as many sources of error as possible, the committee has adopted a new method, which involves the determination of the sound power level directly by comparison with a calibrated standard sound source.[13] This result is accomplished by measuring the octave band sound pressure levels of the calibrated source and of the equipment in the test room. The sound power level of the equipment is then determined directly as the difference of the equipment and standard source pressure levels plus the standard source power level. For instance, if the sound **pressure** level reading of the standard source is 70 db and the reading of the equipment is 50 db, then if the source **power** level is 85 db, the equipment **power** level will be 65 db.

While a variety of sound sources is available for such comparison, the source which is favored by the committee consists of a small, direct motor driven fan wheel with restricted inlet. This fan has the advantage of providing a broad band source of sound of nearly uniform level for all eight frequency bands. At the same time it is small and relatively portable and can be operated on readily available 60 cycle ac current.

One of the most attractive features of such a source is its high degree of stability. The output should remain virtually constant over a long period of time, subject only to mechanical damage or the accumulation of dirt, both of which are readily observable and can be corrected. Minor corrections of the sound power output for barometric pressure, temperature and speed can be provided if desired. The speed can readily be measured by means of a stroboscopic disk and neon light. The sound power output of the standard sound source may vary slightly between a location in a corner or in the middle of the room. However, by calibrating in the position of use, this possible source of error can be eliminated.

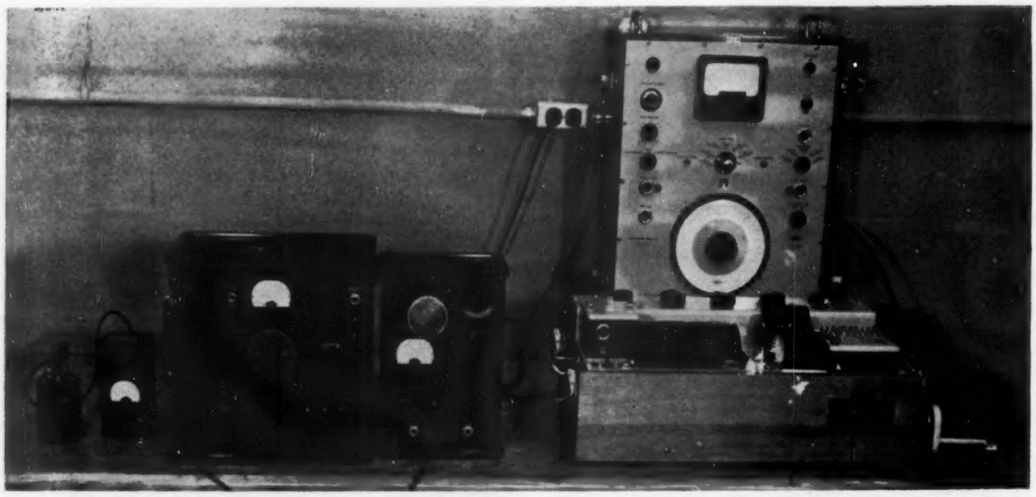
Modified anechoic room suitable for the calibration of the standard sound source. Note also a standard sound source and microphone

PRIMARY STANDARD

It is intended that the primary standard be developed for equipment which will radiate sound to the air in a room or to the atmosphere outside. This category includes a wide variety of equipment ranging from room air conditioners and room fan coil units through self-contained air conditioners, ventilating fans, unit heaters, cooling towers and refrigerating equipment to large refrigerating machines of the reciprocating, centrifugal and absorption types. It is accepted that a single standard can be used for all of this equipment. The primary differences will be in the size of the test space required and the degree of isolation from outside noises. Since the basic standard can provide for such differences, it should be possible to cover the complete range with one standard.

A sub-standard of this standard can provide for the special problems involved in connection with room air conditioners, attic units and the like in which a portion of the equipment radiates to a room and another portion radiates to the outside. The principal additional requirements for this sub-standard are a description of the conditions of mounting and of the treatment of the space surrounding the other part of the equipment. The sound power level will be determined first for one side and then the unit will be reversed and the sound power level determined for the other side.

Fan sound standard – A second standard is needed to cover the sound power level transmitted by fans through ducts. Two general approaches to this problem have been considered. The first is to measure the sound power flux through the ducts by measurements taken in the ducts. The second is to measure the sound power flux from the ducts by measurements taken in a room in which the duct terminates. There are technical problems involved in obtaining a correct result with both of these methods. However, the present thinking of the committee tends to favor the use of the second method because it is more consistent with the procedure for measurement of radiated sound



Measuring facilities for determining the sound power level of air conditioning and refrigerating equipment. Left, microphone calibrating hood and oscillator, octave band analyzer and sound level meter. Right, one-third octave band analyzer driven by a graphic level meter

and because there appear to be fewer problems in connection with this method.

Perhaps the most important of the problems which must be solved is to prevent the low frequency sound from being reflected back through the fan from the duct termination. In order to accomplish this goal, it is proposed to provide the termination with a large horn (expanding section) which will act as a transformer to match the impedance of the duct to the impedance of the room, thus minimizing the amount of reflection.

Other problems have to do with a suitable termination of the duct on the other end of the fan, arrangements for control and measurement of the air quantity and pressure and the measurement of the sound radiated from the fan casing itself. These problems are being considered by the committee and it is hoped to have a tentative standard in draft form in the near future.

Terminal (outlet) sound standard – A third category of equipment is that of duct terminals such as outlets, grilles, induction units and the like. These terminals serve both to attenuate sound and also to generate it due to the flow of air through the terminal. Thus, to describe completely the sound characteristics of the terminal, it is necessary to provide a double set of information.

The amount of sound generated by the terminal can be measured readily by supplying the terminal with a quiet source of air and measuring the sound radiated using the same techniques as those described above for radiation from equipment.

The same set-up can also be used for measuring the amount of sound transmitted from the duct work through the terminal by cutting off the air supply and feeding sound through the duct connected to the terminal by means of a loudspeaker. The sound flux entering the terminal from the speaker can be measured by removing the terminal and substituting for it a horn to act as a transformer so that the impedance of the duct will be coupled properly to the impedance of the room. Using this technique all of the readings will be taken in the same manner and will directly compare the sound power from the equipment with the sound power of the standard source.

FUTURE STEPS

What are the future steps which must be taken to put this series of proposed standards into practice? The next step is a wider circulation of the primary standard in draft form to obtain the comments and criticisms of persons and organizations interested in its use. The primary purpose of this paper is to provide further explanation and background to go with the standard when it is circulated and to explain the background of the standard to our members so that they will be in a better position to understand it, to make suggestions and to use it when it is issued. There may be a place for further discussion at the national or local meetings of the background of the standard and of many of the problems which are involved in connection with the sound testing of equipment. An important part of the proving out of the standard will be the testing of the same piece of equipment in a series of laboratories using the practices recommended in the standard.

After the standards committee has received comments on the draft of the standard, it will be revised again in light of the comments received and will then be ready for formal submission to the standards committee of the Society. Following a review by the standards committee, the proposed standard will then be published in the ASHRAE JOURNAL or otherwise presented to the entire membership for comment. After having received and incorporated the comments received from this step, the standard will then be ready for formal approval. This process will probably take between one and two years.

Cooperation with other groups – In the meantime your committee is working actively on a cooperative basis with other groups. Among these are the committee of the American Standards Association which has written a proposed generic equipment sound testing standard[14] which has been used in large part as a starting point for the present standard by your committee; a committee of the American Institute of Electrical Engineers which is writing a standard for the sound testing of small, medium and large sized motors; and a committee of NEMA which has been designated to write a sound

standard for room air conditioners. The committee is also cooperating with committees which have been set up by trade associations, ARI and AMCA who are working in parallel to prepare practice procedures for the sound rating of air conditioning, heating and refrigerating equipment and the interpretation of the rating in terms of acceptable levels for different applications.

By the time the test standard is completed and approved, it is to be hoped that the work of the trade associations will be advanced to the point where a time table can be set up for putting the sound testing and rating into practice. It is also hoped that arrangements can be made with an independent testing laboratory to provide sound testing service to those manufacturers who do not wish to provide their own facilities.

If the steps being taken obtain general approval, it should then be possible to provide sound ratings of air conditioning and refrigerating equipment. Thus, it should be possible for the first time to provide a clear division of responsibility: to the manufacturers for the sound output of the equipment which they manufacture and to the consulting engineer, architect and owner for other sources of noise and for the proper installation of the air conditioning equipment.

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DISCUSSION at Lake Placid upon the conclusion of the presentation of this paper included remarks from which the following have been excerpted.

NORMAN SAWYER This business of specifying the noise of equipment by total sound power per octave band is a useful concept. It is independent of any environment that you put the equipment in; thus it is a characteristic of the equipment and not of the environment, and you can with that number generally calculate what the sound level will be in different surroundings.

It is also useful for calculating the sound transmitted from duct work which is quite valuable. How would you feel about specifying it in one-third octave bands?

Mr. Ashley Personally, I would be very much in favor of that, but I think the problem of getting the rest of our brothers to swallow that at one fell swoop would be too much.

J. B. CHADDOCK Perhaps some of you in looking at Table I for the first time, feel that the room volumes which were specified are quite large and I think, at first glance, this may seem the case. The Committee spent quite some time experimenting with the room volume problem. These room modes in the reverberant room do lend a problem to accurate measurement. And, in the low frequency bands particularly, it requires a large room to do this. Our choice of 6,000 volume was for the purpose of having 10 modes in the lowest octave band, and therefore taking a relatively few and have a good sound pressure level. It might be well to point out the size of this room.

Mr. Ashley I think Dr. Chaddock has emphasized a notably important point which is likely to be somewhat controversial because some of the companies that are ill-equipped, who already have facilities, will find it is more than they wish to undertake. We feel it is important to have a large test facility to get accurate readings; the present room requirement is minimum room size based on 10 modes in the lowest octave band. That is up to 75 cps and somewhat considerably larger number in the next octave band.

When you consider that the response of the room is greatly peaked and that by increasing the amount of absorbing pressure, you can finally get the peak response down so that it is two-thirds or three-quarters of a cycle width, it means that you are trying to spread three-quarters of a cycle over a total band which would be about 40 cycles and that means you are going to have a gap. In other words, it still isn't as good as we like to have it. We feel it represents a workable compromise.

JAMES P. LAUGHLIN I am working along this idea of 6,000 volume and realize many of us in the industry have a pretty limited capacity generally in the current sound room. Also, there is a general premium on factory space with generally a high noise environment. Will the industry as a whole tend to support the need for the larger room or, again, is there a probability that by using some type of adapting system or perhaps some conversion factors that we might still utilize some of the existing rooms?

Mr. Ashley Unfortunately, the difficulty with the smaller rooms is that the sound may be either amplified or greatly diminished depending upon the particular characteristics of the room; that is, the distribution of the room mode in relation to the single existing frequency. We had an earlier so-called dead room in which we happened to have a frequency, if I am not mistaken, that peaked at 120 cycles and we got a terrific response from that; the reverse can happen. In other words, if you happen to hit a valley between room mode response, then the sound which should be measured accurately, will not be measured and the difficulty is that there isn't any way of correcting for that. You have to accept it as an inaccuracy of the method.

E. H. JENSON I would like to say that we have a room which is nearly the minimum size that you describe and we do have the problems that you have described. I know that the minimum that you have may be a bit too small. The ceiling we have is substantially lower than what you have and that may make a difference. We have about an eight foot ceiling, so I think, if anything, the minimum should be higher than it is.

Mr. Ashley Well, I am gratified to see that we have some support on the other side. I think you are probably right from a technical point of view. I think that the—well, my own feeling is that about 10,000 cu ft is about as small as the room should be to bring good results technically. But, I think also we have to recognize that we are still in the area of compromise and the feeling of our Committee was that the 6,000 cu ft represents a not too bad compromise.

Now in saying that, I think that we are thinking partly in terms of the measurement of broad band noise. Broad band noise is good, no question about that. For single frequency noise measurement, it is distinctly a compromise and I think the best we can say about it, there is some need for some research to improve the diffusion of sound over the whole frequency spectrum. And I am quite confident, as a matter of fact, that we can do that. In other words, we are thinking in terms of additional features as part of the room that will improve the diffusion.

One of the methods which has been proposed and which is used in a number of laboratories is the revolving vane. It

(Continued on page 114)

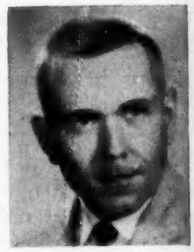
Optimum conditions for Fresh food preservation in the domestic refrigerator

Studies of fresh food preservation have been published both from the standpoint of food technology and from the standpoint of refrigerator design. These areas of activity differ; the food technologist explores preservation factors while the design engineer studies related functional embodiments.

The science of food technology applies recognized methods of evaluating microbial growth and chemical changes as a function of temperature and humidity. These tests are controlled and measured by techniques appropriate only to the laboratory. Present studies indicate the existence of a temperature and humidity focal point for optimum food storage.

Design engineering, on the other hand, involves an end product – the domestic refrigerator. Collectively, refrigerators exhibit a wide temperature and humidity spectrum and a variety of basic design features. Engineering-wise, a given design can be evaluated only in the light of existing standards.

E. W. ZEARFOSS
Member ASHRAE



F. P. SPEICHER

FOOD TECHNOLOGY

To perform its major function, the domestic refrigerator should provide optimum storage conditions for the extended preservation of a variety of perishable foods. These foods include such diverse groups as fresh meats, vegetables, fruits, dairy products and leftovers.

In defining optimum storage conditions, it is important to distinguish between conditions desirable for ripening, aging and maturing after harvest or production, and those which subsequently are best for preserving or keeping the foods from deterioration. Also, it should be recognized that fresh foods may have a questionable history when marketed to the homemaker, and perhaps are rapidly losing their quality and approaching the end of their edible life. In any event, the optimum conditions for fresh food preservation in the household refrigerator are those which affect maximum edible life of the stored foodstuffs commensurate with their market day quality.

To extend the storage life of a food, the chemical and microbial changes responsible for deterioration must be retarded. These changes are minimized by low temperatures. To illustrate the relationship between microbial growth rate and refrigeration temperatures, petri dishes filled with sterile nutrient agar were streaked with species of food-destroying microorganisms and incubated in both saturated (100% r.h.) and dry (40% r.h.) air at temperatures of 45, 39 and 33 F, respectively, for seven days. The micro-organisms used were (1) a mixed culture of bacteria which had been isolated from spoiled refrigerated beef, (2) a pure culture of bacteria from spoiled peas, and (3) a mold isolated from fresh vegetables. Photographs of the dishes were taken on the third, fifth and seventh day of incubation and are presented in composite form in Fig. 1.

These data show that reducing the refrigeration temperatures to the near-freezing point results in a significant decrease in microbial growth, and indicate that fresh foods stored at this temperature would exhibit minimum microbial spoilage. The identical growth rates of the bacteria at 40% and 100% r.h. when incubated at the same temperature show that temperature is the dominating factor. This suggests that one shortcoming usually associated with high humidity, the sliming of meats, can be curtailed effectively by near-freezing temperatures.

Another benefit of lower temperatures is the slowing down of the natural chemical changes which result in browning, softening, etc., of many fresh foods. This is well substantiated by research reports from many food laboratories which reveal that meats and most fruits and vegetables keep better and

F. P. Speicher is a Biologist and E. W. Zearfoss is Development Engineer, Advanced Development Dept, Appliance Engineering, Philco Corporation.

This paper was presented at the Domestic Refrigerator Engineering Conference, at the ASHRAE annual meeting, Lake Placid, N. Y., June 22-24, 1959.

longer at near-freezing temperatures.

Equally important in the degradation of refrigerated foods is desiccation. The problem of foods drying and shriveling has been recognized for some time by the homemaker and is a common source for complaints about the household refrigerator.

Although studies of fresh food preservation have been made in the areas of food technology and refrigerator design, a need exists for an index or standard (comparable with the accepted zero degree specification for frozen foods) to guide the industry with relation to fresh food preservation. The authors here discuss the concept of an optimum storage condition, allied with today's trends and capabilities in refrigerator design.

The rate at which a specific food will lose moisture depends on its surface characteristics and on the temperature, humidity and movement of the ambient air. To show the effects of these factors, the dehydration rate of freshly opened peas was determined at various relative humidities and temperatures in still air using the following procedure.

The desired relative humidity was maintained in a 4 x 8 x 11 in. sealed plastics container by means of a solution of water and glycerine. Items placed in the chamber were set on a metal trivet above the surface of the solution. A thermocouple and an electric hygrometer sensing element were attached to the inner surface of the lid (Fig. 2). Four such chambers containing glycerine-water solutions calculated to give relative humidities of 100, 80, 60 and 40%, respectively, were placed in a constant temperature refrigerated cabinet. A petri dish containing approximately 50 grams of pre-chilled canned peas was weighed and placed in each chamber. After three days the dishes were again weighed and the weight loss per 24 hr was computed for each dish. Tests were conducted in this manner at temperatures of 45, 39 and 33 F. Results of these tests (see Fig. 3) show the recognized relationship between evaporation rate, temperature and relative humidity.

To simulate the effects of natural convection currents in a conventional refrigerator, a small 600 rpm fan with a 2-in. diam blade was mounted directly above the peas with the motor external to the chamber. The 24 hr weight loss was determined for various relative humidities at 33 F. These data are compared with 33 F still air evaporation rates in Fig. 4. The curves illustrate the considerable increase in evaporation rate caused by a relatively small air movement.

It should be noted that the rate at which a specific food item loses moisture to the atmosphere also depends upon its exposed surface area and the nature of its surface. Thus, various foods show different weight losses per 24 hr under the same conditions. Canned peas were selected for these tests for reasons of convenience and repeatability.

Based on the results of these studies and related reports it may be presumed that the optimum storage conditions for preservation of most foods in the household refrigerator are at the focal point of a near-freezing temperature, high humidity and still air. This can be illustrated graphically as shown in Fig. 5.

DESIGN ENGINEERING

The engineering achievement of the optimum environment derived in the foregoing discussion of food technology can now be described.

When the surface of a hollow geometrical configuration exists at a uniform temperature, the enclosed volume will reflect this temperature. Further, moist foodstuffs within this volume will produce a saturated vapor-air mixture. Any subsequent temperature depression

Fig. 1 Effects of temperature and humidity on the growth rate of food micro-organisms (bacteria and mold)

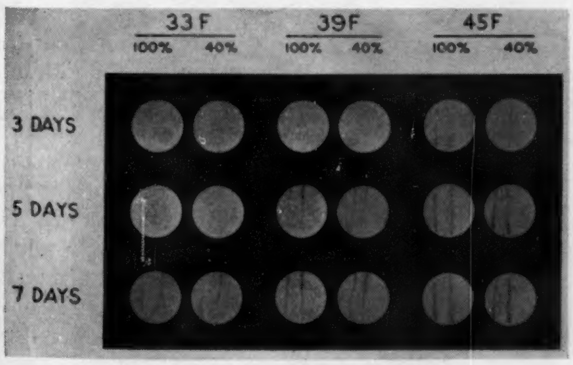
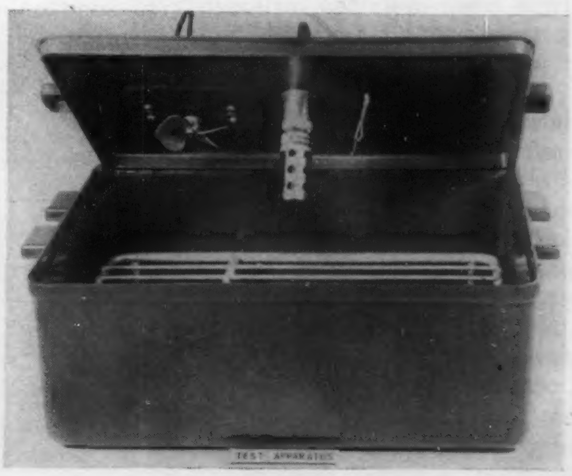


Fig. 2 Test apparatus



on a given boundary area causes water vapor to condense from the mixture thereby decreasing the original vapor pressure and humidity. Correspondingly, the vapor pressure differential effected between foodstuffs and the unsaturated air comprises the mechanism for food desiccation or dehydration. Dehydration rate depends upon the magnitude of the area and temperature deviations noted above. Further, food dehydration will occur when the compartment is not vapor-tight. Basically, these parameters define the engineering problems involved in producing and controlling a saturated vapor-air mixture in a compartmented section of a refrigerator.

Refrigerators have been compartmented for many years. Sometimes the compartment did little more than provide convenient storage for one kind or group of foodstuffs. Individual compartments have thus accommodated vegetables, fruits, meats and dairy products. In some instances the design temperature or humidity was expressed and advertised in such relative terms as high humidity or low temperature. Compartment designations or trade names frequently included an environmental connotation to support this theme. In any case, these compartments for the most part had various psychrometric properties.

Although there is now a trend toward marketing features which emphasize food preservation, no domestic refrigerator today has a single compartment designed to combine and store adequately the diverse groups of foodstuffs that benefit from controlled temperature and humidity. However, the optimum conditions for a wide variety of foods described earlier in this paper suggests a new philosophy for modern design.

Contemporary engineering trends and developments in domestic refrigerators cooperate to allow this objective to be realized. One embodiment which achieves this objective is shown in Fig. 6. A vapor-sealed compartment having a separate closure is provided within the cabinet liner. Cabinet air is drawn into a duct system over an evaporator. The refrigerated air is then divided; one part is discharged to the upper regions of the

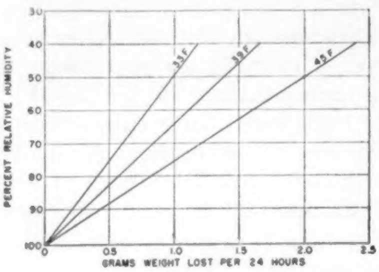


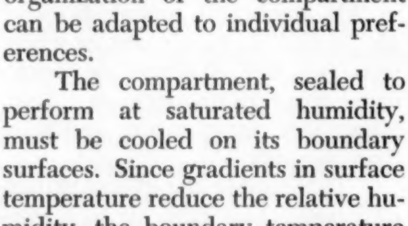
Fig. 3 Effects of temperature and percent relative humidity on desiccation of peas

cabinet, while the second part flows downward through the duct toward the lower portion of the cabinet. Arrows on the sketch show that four vertical walls of the sealed compartment are literally wrapped in cold air. When an insulated freezer section is located below the fresh food compartment, passage of air beneath the compartment is optional. The air system, powered by a small motor and blower assembly, has a thermostatic control. Dimension-wise the compartment readily accommodates assorted vegetables, fruits, cold cuts, meats, leftovers, etc., all stored uncovered. The functional organization of the compartment can be adapted to individual preferences.

The compartment, sealed to perform at saturated humidity, must be cooled on its boundary surfaces. Since gradients in surface temperature reduce the relative humidity, the boundary temperature must be as uniform as good design can achieve. Thermally conductive materials, especially in the compartment vertical wall sections, help to achieve this objective.

A forced convection system will allow a higher rate of air circulation than a gravity system, and a correspondingly decreased temperature differential measured around the complete air circuit. This minimizes compartment surface temperature gradients and affords a marked improvement in overall design. In using forced air the location of the compartment within the cabinet might appear flexible to the whims of utility and styling. However, the lower section presents definite advantages since air stratification allows cabinet regions above the sealed volume to be controlled at a temperature higher than that of the compartment. This warmer zone is important for several reasons yet to be noted.

Fig. 4 Effect of moving air on desiccation of peas at 33F



The advantages of forced convection to the design of a low temperature, high humidity compartment are:

1 – Following cabinet usage, forced convection provides quick recovery to the optimum temperature. This is an important factor in actual food storage life. Parenthetically, the separate closure on the compartment likewise minimizes cabinet usage effects.

2 – Moisture deposited on surfaces exterior to the sealed compartment during cabinet usage is evaporated readily by forced convection.

3 – To avoid freezing of the foodstuffs, temperature variations in the critical near-freezing zone must be minimized. Forced air, controlled by a reliable thermostat, helps solve this tolerance problem.

4 – Compartment surface temperature gradients must be minimized. Forced air directed to these surfaces achieves this objective.

5 – Creation of warmer zones in other cabinet regions may be desirable. A forced air system can be divided to produce this higher temperature zone in the the upper part of the cabinet while maintaining near-freezing temperatures within the compartment.

As stated previously, surface temperature gradients have an undesirable effect on the compartment humidity. There is, however, another more subtle and significant thermal mechanism stemming from the gradients. Convection currents may be induced with-

in the compartment by uneven temperature distribution on the compartment boundary, especially if the top surface temperature is depressed. Convection currents within the compartment supplement the slower process of pure vapor pressure diffusion and have a marked effect on food desiccation, as shown in the preceding section on Food Technology. Consequently, the migration of moisture from the foodstuffs to the interior surface of the compartment is doubly dependent upon the temperature pattern on the compartment boundary.

Positioning the sealed compartment below a higher temperature region reverses the direction of heat flow through the compartment top surface. This arrangement, by increasing the top surface temperature, curtails the development of convection currents and obviates moisture accumulation and drippage to the compartment contents.

Any condensation on this surface during usage of the compartment facility will subsequently migrate to cooler wall surfaces within the compartment. Moisture on these surfaces presents no drippage problem.

Temperatures in the 36 to 40 F range are necessary for acceptable natural or cyclic defrost of the evaporator in the fresh food section of the cabinet. This higher temperature zone, external to the sealed compartment, fortunately coincides with the needs of other groups of foodstuffs, and can be compartmented for utility and styling or left with open shelves. These

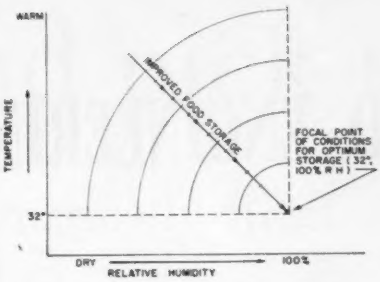
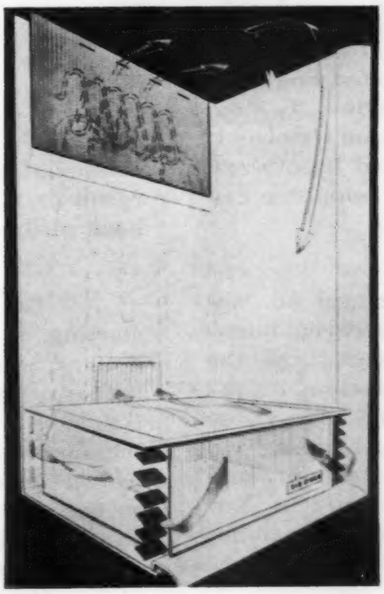


Fig. 5 Effects of temperature and humidity on storage life of perishable foods

Fig. 6 Schematic arrangement of compartment designed for optimum storage conditions for extended preservation of fresh foods



regions will provide satisfactory storage for canned foods, bottled goods and beverages, dairy products and other foods not readily desiccated or deteriorated by higher temperatures.

CONCLUSION

Actual food preservation tests were conducted on a refrigerator equipped with a special compartment designed to provide optimum storage conditions, as shown on Fig. 6. The compartment was loaded with a cross-section of foods especially responsive to temperature and humidity. These included, all uncovered: Meats — chicken, pork chops, hamburger, steak and lunch meats; Vegetables — celery, lettuce, tomatoes, parsley; Leftovers — mashed potatoes, whipped cream, a sandwich, sliced tomatoes and chocolate pudding. Using a group of current model refrigerators on the domestic market, similar foods were placed in the recommended storage region or facility. Observations were made on all foods every day by a test panel that included a Home Economist.

The optimum temperature and humidity air-wrapped compartment provided excellent preservation of all food specimens, both from a quality and a time duration standpoint. These tests confirmed the premise that a near-freezing temperature, a high humidity and still air are optimum for fresh food preservation.

The authors express their appreciation to Lloyd A. Staebler, Manager, Advanced Development, for his guidance and encouragement in the preparation of this paper, and to Adelaide Fellows, Director, Home Economics, for her help in conducting and evaluating the food load tests.

HEATING-COOLING ELECTRICALLY ON RIVER TUG

Baseboard electric heating and small air conditioning units are provided in each of the six cabins on the "Captain J. G. Van Ness." The wheelhouse is also electrically heated: the galley is weather conditioned with electric heating and air conditioning, too.

Baseboard electric heating was chosen for its fast warm-up characteristics and ease of individual thermostatic controls. Here shown is the wheelhouse with skipper examining the controls and the Westinghouse baseboard electric heating.

NEW UNITED ENGINEERING CENTER

MERRILL F. BLANKIN

After four years of intensive effort by the founder societies and associated engineering societies, the eighteen story United Engineering Center is under construction. Ground breaking ceremonies took place on October 1, 1959 with ASHRAE represented by officers, directors, UEC Fund Raising Committee and members.

The Center is being erected on New York's United Nations Plaza between Forty-Seventh and Forty-Eighth Streets. On the facing page is a color reproduction of the architect's perspective drawing. Truly for many, many years to come this building will bear witness to the high ideals, stability and prestige of all engineers. Not only in New York, but across this continent and around the world.

We, of ASHRAE, have an important role to play in the operation and construction of the Center. Our headquarters will be combined and occupy over 4,000 square feet in the building when it is opened in the Spring of 1961. We are now asked to help defray part of the total construction costs, which will be in excess of ten million dollars.

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The founder societies have had their campaign underway for some time and as a result over seven million dollars has already been pledged.

The goal for our campaign, just starting, is two hundred and fifty thousand dollars; which may seem like a great deal of money. Actually, each individual subscription would be extremely modest if every member would do his share, approximately fifteen dollars per member, with payment spread over three years if desired. Also, the contribution is tax deductible. However, since universal support never seems possible, engineers must be counted on to give according to their abilities, which vary according to the rewards that engineering has afforded them.

The principal requirement is for every engineer to recognize his responsibility and act upon it. The pride and prestige that his immediate financial support will insure will be enhanced for many years to come by the building itself.

For your convenience a contribution pledge form is incorporated within this page. Mail to ASHRAE, 62 Worth Street, New York 13, N. Y.

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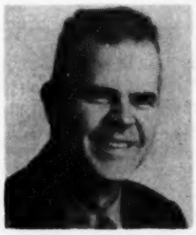
Piping and pick-up factors

for automatically fired one-pipe, hot-water and steam systems



WARREN S. HARRIS
Member ASHRAE

ROBERT R. LASCHOBER

CHIEN FAN

In designing hot-water or steam systems, it is common practice to select a boiler having a gross output in excess of the actual load to be connected to the boiler. This is to insure that the boiler will have sufficient capacity to take care of the piping loss of the system and to bring the system up to its maximum output within a reasonable time following a period of idleness. The ratio of the gross boiler output to the total connected load is called the piping and pick-up factor. The piping and pick-up factors used in past years for automatically fired boilers have ranged from about 1.25 to as high as 1.56. In the interest of economical installations, it is desirable to use as low a factor as possible consistent with proper performance of the system.

The investigation reported in this paper was undertaken to supply data concerning the effect of the size of the piping and pick-up factor on the performance of residential hot-water and steam heating systems and to provide a logical basis for determining the minimum piping and pick-up factor which should be used.

The experimental work on hot-water systems was done in the I-B-R Research Home. This investigation* was carried on in the Dept. of Mechanical Engineering at the University of Illinois as a part of the research program on hydronic systems sponsored by the Institute of Boiler and Radiator Manufacturers.

Warren S. Harris is Research Professor, Chien Fan is Research Assistant and Robert R. Laschober is Research Associate, Dept of Mechanical Engineering, University of Illinois.

I-B-R RESEARCH HOME

The Research Home, described in detail previously,[1,2,3] was a two-story building typical of the small, well-built American home of 1940. The construction was brick veneer on wood frame. All of the outside walls and the second-story ceiling were insulated with mineral wool batts 3⅝ in. thick. A vapor barrier between the plaster base and the insulation prevented condensation on the sheathing by retarding the passage of water vapor from the rooms into the insulation in the walls. The calculated coefficient of heat transmission, U, for the wall section was 0.074 Btu per sq ft per hr (F) temperature difference. All windows and the two outside doors were weatherstripped. Two storm doors were used. The total calculated heat loss for the house, excluding the basement, was 43,370 Btuh at design temperatures of –10 F outdoors and 70 F indoors. A summary of room volumes and calculated heat losses is given in Table I.

* A complete report of the investigation is being published as Univ. of Ill. Eng. Exp. Sta. Bul. No. 455.

HEATING SYSTEMS

1950-51 Hot-Water System – A gas fired, one-pipe forced circulation, hot-water system, designed in accordance with I-B-R Installation Guide No. 5, was used in the Research Home during the 1950-51 heating season. The capacity of the installed radiation was equal to 40,190 Btuh at a design water temperature of 215 F. This capacity was equivalent to the calculated heat loss of the house at an indoor-outdoor temperature difference of 74 F or to a design outdoor tem-

Exploring that margin by which boiler gross output capacity exceeds actual load connected to the boiler, the authors have derived test data applicable primarily to residential heating systems. This study is the 15th in a closely related series.

TABLE I——DATA ON I-B-R RESEARCH HOME & HEATING SYSTEMS

				1950-51 Hot-Water System			1956-57 Hot-Water System			1957-58 Steam System		
				INSTALLED RADIATION			INSTALLED RADIATION			INSTALLED RADIATION (23" 3-Tube, Large Tube)		
ROOMS	DIMENSIONS	HEATED SPACE CU FT*	CALC. HEAT LOSS BTUH	NO. OF UNITS	QUANTITY	RADI-ATION OUTPUT BTUH	NO. OF UNITS	QUANTITY	RADI-ATION OUTPUT BTUH	NO. OF UNITS	NO. OF SECT.	RADI-ATION OUTPUT BTUH
	FIRST FLOOR CEILING HEIGHT 8'3"											
Liv R	24'0" x 13'4"	2641	5749	1	11 ft RC††	5335	1	13 ft RC††	6305	2	13	6240
Din R	13'1" x 11'3"	1183	8742	1	11 ft RC††	5335	1	13 ft RC††	6305	1	14	6720
Kitchen	10'5" x 11'3"	799	3199	1	10 Sect ST†††	3360	1	11 Sect ST†††	3696	1	7	3360
Lavatory	7'0" x 2'8"	152	1484	1	8 Sect ST†††	2688	1	9 Sect ST†††	3025	1	2	960
Vestibule	7'5" x 5'4"	284	4848	1	14 Sect ST†††	4704	1	16 Sect ST†††	5376	1	8	3840
Vest Closet		54										
TOTAL 1st Fl		5113	24022	5		21422	5		24707	6	44	21120
	SECOND FLOOR CEILING HEIGHT 7'9"											
NE Bed R	10'7" x 9'9"	800	4393	2	9 ft RC††	4365	2	10 ft RC††	4850	1	9	4320
NW Bed R	10'6" x 13'4"	1148	4944	2	10 ft RC††	4850	2	12 ft RC††	5820	1	11	5280
SW Bed R	13'0" x 11'4"	1108	5250	2	10 ft RC††	4850	2	11 ft RC††	5335	1	11	5280
Bath R	6'6" x 7'6"	374	2606	1	8 Sect ST†††	2688	1	9 Sect ST†††	3025	1	5	2400
Stairway		505	2155	1	6 Sect ST†††	2015	1	7 Sect ST†††	2353	1	3	1440
Closets		345										
TOTAL 2nd Fl		4280	19348	8		18768	8		21383	5	39	18720
TOTAL 1st & 2nd Fl		9393	43370	13	51 Ft & 46 Sect	40190**	13	59 Ft & 52 Sect	46090***	11	83	39840†

* No Storm Sash — Outdoor Temperature = —10 F. Indoor Temperature = 70 F. Infiltration Based On Crackage.
** Equivalent to a design temperature difference of 74 F or a design O.D.T. of —4 F.
*** Equivalent to a design temperature difference of 85 F or a design O.D.T. of 15 F.
† Equivalent to a design temperature difference of 73.5 F or a design O.D.T. of —3.5 F.
†† 7" RC Radiant Baseboard
††† 19" 4-Tube Small Tube

perature of –4 F and a design indoor temperature of 70 F. The radiation (see Table I) consisted of five units of 19-in. 4-tube, small tube radiators, totaling 46 sections, and eight units of 7-in. type RC radiant baseboard totaling 51 ft long. All radiation was located along outside walls and under windows in so far as was possible. The bulk of the piping system was located in the basement.

The wet bottom, cast iron boiler composed of two 6-in. sections and one 4-in. section was located in the basement. This boiler was insulated on top, sides and back with an air cell insulation approximately 1 in. thick and was completely enclosed in an enameled sheet metal jacket. All cracks between sections were sealed with asbestos cement. The boiler was equipped with a conversion type gas burner. The net I-B-R rating of the boiler was 55,000 Btuh, the gross I-B-R output was 84,000 Btuh; however, during the tests reported here several different firing rates were used in order to obtain different gross outputs.

1956-57 Hot-Water Systems – The hot-water system used during the 1956-57 heating season was the same as that used during 1950-51 except that the installed radiation (see Table I) was increased by about 12%, equivalent to using a design indoor-outdoor temperature difference of 85 F.

1957-58 Steam System–A gas fired, one-pipe steam heating system was used in the Research Home for all tests made during the 1957-58 heating season. The capacity of the installed radiation (see Table I) was equivalent to the calculated heat loss of the house at an indoor-outdoor temperature difference of 73.5 F. The radiation consisted of 23-in. 3-tube, large tube radiators set under windows. All radiator venting valves and the main vent valves were of the non-vacuum type. The venting rates of the valves were adjustable. A 2-in. main in the basement conveyed steam to the radiators and returned the condensate to the bottom of the boiler.

A dry base, 4-section, cast iron steam boiler designed for gas firing was used in the tests. The boiler was insulated on the top, sides, front and back with glass wool insulation approximately 1 in. thick and was completely enclosed in an enameled sheet metal jacket. The net I-B-R rating was 40,000 Btuh, but, as was the case with the hot water boiler, several gas burning rates were used to obtain a range of gross outputs.

CONTROLS

The same basic control arrangement was used in all three heating systems. The room thermostat, located at the 30-in. level in the living room, was of the heat-anticipating type and turned the gas burner on and off according to the heat requirements of the room. On the hot-water system the room thermostat also controlled the operation of the circulator. The safety control consisted of an immersion thermostat set to stop the burner, but not the circulator, when the temperature of the water in the boiler exceeded 225 F. The burner would re-start when the water temperature dropped to about 185 F.

The safety controls on the steam system consisted of a low water cut-off, set to turn off the burner any time the water level in the boiler dropped 5 in. below nor-

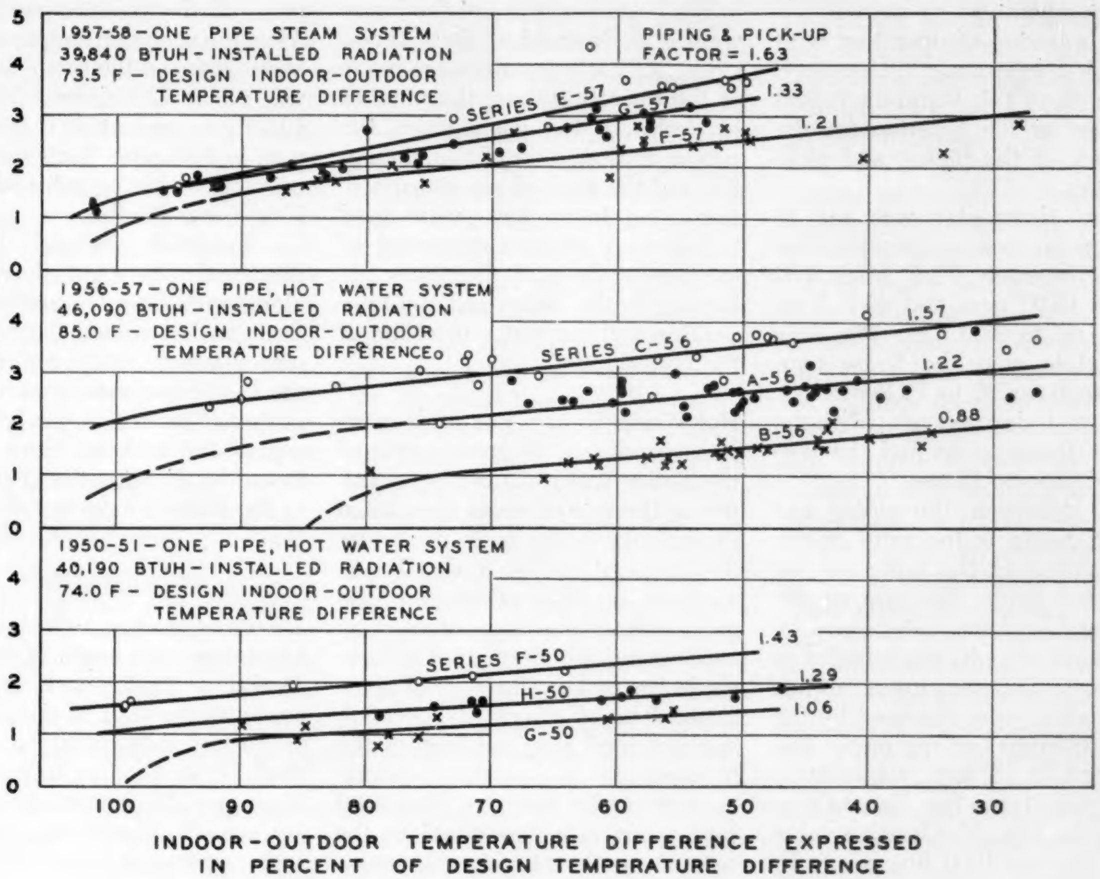


Fig. 1 Effect of design temperature difference on the warm-up rate

mal; and a steam pressure control, set to stop the burner when the steam pressure attained 2.5 psi gage.

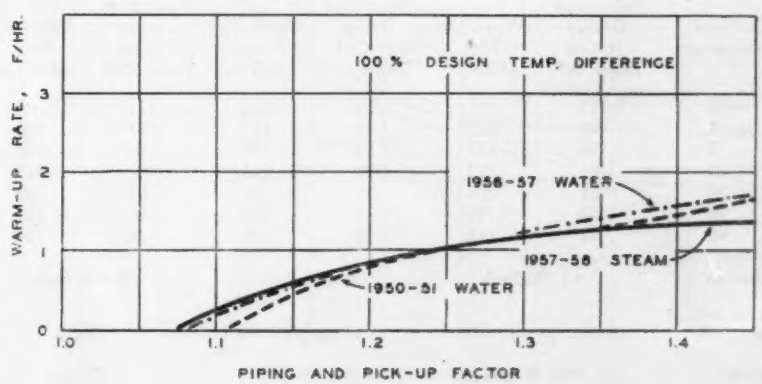
Both the steam and water boilers were equipped with safety valves set to open at the maximum allowable working pressure for the system.

TESTING APPARATUS

Thermocouples for the measurement of room-air and basement-air temperatures were installed at levels of 3, 30, and 60 in. above the floor as well as 3 in. below the ceiling. Other thermocouples were used to measure the temperature of the water or steam leaving the boiler and the temperature of the water returning to the boiler. A differential pressure recorder attached to an elbow meter,[4] calibrated in place, supplied a continuous record of the rate of water circulation through the boiler for all tests with the hot water system.

Recording thermometers were used to make continuous records of the air temperatures in each of the six rooms and the attic space at 3 and 30 in. above the floor and 3 in. below the ceiling. Similar thermometers recorded the temperature of the outdoor air and the temperature of the flue gases at the flue outlet of the boiler. The CO_2 content of the flue gas was obtained by use of an Orsat apparatus, and the fuel consumption was measured by a dry test meter. The moisture content of the air was measured by four humidity indicators and one hygrometer. A thermocouple-type surface pyrometer was used to measure the temperature of radiator surfaces during special studies.

Fig. 2 Effect of piping and pick-up factor on the warm-up rate



TEST PROCEDURES

General Method – Operating a heating system with reduced temperature at night is one of the more common methods of operation requiring an adequate piping and pick-up factor. The smaller the factor used, the longer the time required for the system to raise the

house temperature to normal following a period of operation with reduced temperature. Therefore, the length of this warm-up period was used as the criterion for the adequacy of the piping and pick-up factor.

The thermostat was set to maintain a room-air temperature of approximately 72 F from 5:30 a.m. to 10:00 p.m. and 66 F from 10:00 p.m. to 5:30 a.m. The time required to raise the house temperature from 66 to 72 F was observed each day and related to the outdoor temperature and the piping and pick-up factor.

By definition, the piping and pick-up factor is the ratio of the gross output of the boiler to the connected load. The size of the connected load was determined by the amount of radiation installed in the house. Different gross outputs of the boiler were obtained by the use of different burning rates. The fuel used in all tests was natural gas supplied by the Texas-Oklahoma pipe line. The heating value of the gas was 1000 Btu per cu ft.

At all times during the tests the doors between rooms were open, and the windows were closed. Observations of room-air temperature as determined by the thermocouples located 3, 30 and 60 in. above the floor and 3 in. below the ceiling were recorded at 7:30 a.m., 11:00 a.m., 4:00 p.m. and 10:30 p.m. The temperature of the air in the basement and in the attic, and the relative humidity in the heated portions of the house were also observed at these times. Complete daily records were made of the operating time, the number of cycles of the gas burner, the power consumption of the circulator, and the cu ft of gas consumed. Recording instruments were used to obtain a continuous record of the temperature of the water returning to the boiler and the temperatures of the water or steam at the boiler outlet.

Hot-Water Tests – For each operating condition the gross output of the boiler was obtained by multiplying the rate of water circulation through the boiler in lb per hr by the temperature rise in deg F. As a check on this calculation, the gross output was also obtained by multiplying the heat input rate to the boiler by the efficiency of operation. The efficiency of operation was obtained from readings of the temperature and carbon dioxide content of the flue gas. The total heat input rate was equal to the rate of gas burning in cfh multiplied by the heat content of the gas in Btu per cu ft.

The piping and pick-up factors were obtained by dividing the gross output of the boiler by the connected load, both expressed in Btuh.

Steam System – In testing the steam system the initial steps were to establish a uniform rate of heating rooms by adjusting the venting rates of the radiator vent valves. After the room-air temperature had been balanced the room thermostat was set at 72 F from 5:30 a.m. to 10:00 p.m. and at 66 F from 10:00 p.m. to 5:30 a.m. Tests were made using gas burning rates of 65 cfh, 70 cfh and 85 cfh.

For each series of tests the measured flue gas temperature along with the CO_2 content of the flue gas determined the boiler efficiency which, when multiplied by the heat input rate, gave the gross output of the boiler. As before, the piping and pick-up factor was obtained by dividing the gross output of the boiler by the installed radiation.

RESULTS

Effect of Indirect Water Heater Operation on Length of Warm-Up Period – Preliminary tests were made at the start of the hot-water studies to determine the effect of the operation of an indirect water heater on the length of the warm-up period. Conditions considered were: operation with the low limit control and with no water heater in use; operation with the water heater attached but no hot water used; and operation with a daily hot water consumption of 75 gal.

When the outdoor temperature was near the design temperature, the method of operation had no measurable effect on the length of the warm-up period. In mild weather, operation with no water heater attached to the boiler resulted in the longest warm-up time.

The piping and pick-up factor is most critical when the system is operating at or near design conditions. This being the case, these preliminary tests indicated that the method of operation should have no effect on the results of the tests, however, practically all subsequent tests to determine the minimum piping and pick-up factor were made by operating the boiler with no low limit control and without an indirect water heater attached.

Effect of Percent of Design Indoor-Outdoor Temperature Difference on Warm-Up Rate – To place all data on a common basis, the average rate of warming the room air in deg F per hour during the warm-up period was correlated with the observed indoor-outdoor

TABLE II——EFFECTS OF OUTDOOR TEMPERATURE AND SYSTEM SIZE ON LENGTH OF WARM-UP TIME

Indoor Temperature = 70 F
Piping and Pick-Up Factor = 1.3
House Temperature Raised 6 F

	1950-51		1956-57		1957-58	
Outdoor Temperature, F	Percent of Design In-Out Temp. Diff.	Warm-Up Time, Minutes	Percent of Design In-Out Temp. Diff.	Warm-Up Time, Minutes	Percent of Design In-Out Temp. Diff.	Warm-Up Time, Minutes
—20	122	Inf.	106	500	121	Inf.
—10	108	435	94	225	107	510
0	95	272	83	172	94	233
10	81	227	71	146	81	167
20	68	204	59	130	67	133
30	54	195	47	125	54	115
40	41	195	35	125	40	105
Installed Radiation	40,190 Btuh		46,090 Btuh		39,840 Btuh	
Design In-Out Temp. Diff.	74.0 F		85.0 F		73.5 F	
System	Hot-Water		Hot-Water		Steam	

temperature difference expressed in percent of the design temperature difference, and with the piping and pick-up factor. The design temperature difference was taken as that at which the calculated heat loss of the house would be equal to the capacity of the installed radiation. The observed indoor-outdoor temperature difference was assumed equal to the difference between the average daytime indoor air temperature and the average outdoor air temperature during the warm-up period.[2,5] The piping and pick-up factor was obtained from the heat input rate during the test and the procedure outlined under Test Procedure.

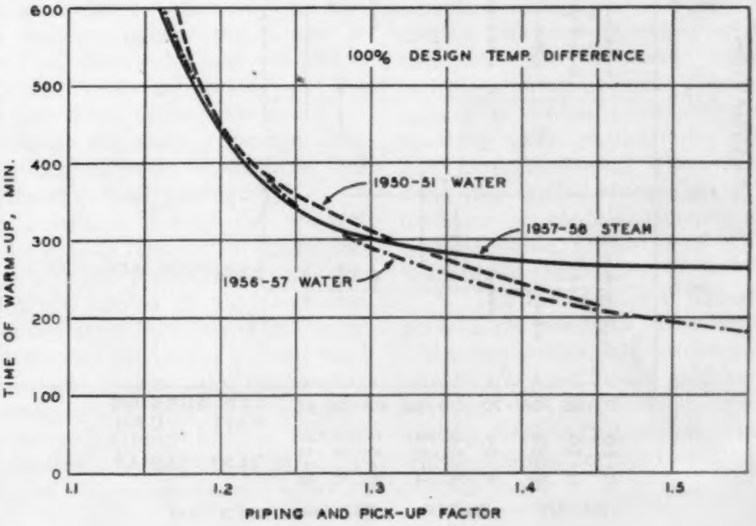
Fig. 1 is a plot of all test data using the method just described. These curves show that for any piping and pick-up factor the rate of warm-up decreases as the indoor-outdoor temperature difference increases and at any given indoor-outdoor temperature difference, the rate of warm-up increases as the piping and pick-up factor is increased. It appeared that the indoor-outdoor temperature difference had a greater effect on the warm-up rate when using the steam system than when using the hot water system. Also, as the size of the hot-water system was increased, the effect of the indoor-outdoor temperature difference on the warm-up rate became more pronounced.

Effect of Piping and Pick-Up Factor on Time of Warm-Up – The warm-up rate at the design indoor-outdoor temperature difference is most important since the boiler size must be adequate to ensure satisfactory operation at these conditions. Fig. 2 shows the relationship between the warm-up rate and the piping and pick-up factor for each of the three systems when operated at design conditions. These curves were derived directly from those in Fig. 1.

It was found that for each piping and pick-up factor the rate of warming up the room air when operating at design indoor-outdoor temperature difference was essentially the same regardless of the type or size of the system. At a piping and pick-up factor of approximately 1.1, the heating capacity of the system was not sufficient to raise the room-air temperature at the end of a night set-back period. Thus a factor of 1.1 might be used on a system where the room-air temperature was to be maintained at a constant value at all times but it would not be adequate if the room-air temperature were to be periodically reduced below the normal operating value. A clearer picture of the effect of the size of the piping and pick-up factor on the length of the warm-up time is obtained from Fig. 3 which shows the time required to raise the room-air temperature 6 F using warm-up rates obtained from Fig. 2. As the piping and pick-up factor was reduced below about 1.3 there was a marked increase in the length of the warm-up time. On the other hand, as the piping and pick-up factor was increased above 1.3, the decrease in warm-up time was relatively small. For example, increasing the piping and pick-up factor from 1.3 to 1.56 (a 12% increase in the gross boiler output) made almost no reduction in the length of the warm-up time when using the steam system, and about a 30% reduction in the case of the hot-water system. A 12% reduction in the gross output of either the steam or hot-water boiler resulted in a warm-up time of over 10 hr.

The results of these tests indicate that a piping and pick-up factor of 1.3 is adequate for the successful operation of both automatically fired steam and hot-water residential heating systems. The use of larger factors results in the selection of larger boilers but offers little or no improvement in overall performance.

Fig. 3 Effect of piping and pick-up factor on 6 F warm-up



Effects of Outdoor Temperature and System Size on Time of Warm-Up – The length of the warm-up period shown in Fig. 3 may seem large. However, this is for design outdoor temperature difference and, for a greater part of the winter, outdoor temperatures are well above design. In Urbana the average outdoor temperature for the winter is 38 F and for about one day out of every two years the outdoor temperature averages zero or less. Furthermore, due to the radiant effect of the hot radiators and the warmer than normal ceiling, the house felt comfortably warm 60 to 90 min before the actual end of the pick-up cycle.

Table II has been developed from the same data as Figs. 2 and 3 and is arranged to show the effect of outdoor temperature and system size on warm-up time. For all systems, an increase in the outdoor temperature resulted in a marked decrease in warm-up time. This was especially true in the 1957-58 tests using the one-pipe steam system. Whereas, for all systems, the length of the warm-up period at design indoor-outdoor temperature difference was about 300 min (Fig. 3), at an outdoor temperature of 30 F it was only 115 to 195 min.

A hot-water heating system inherently requires a relatively long time to raise the house tem-

perature from one value to a higher one.

By increasing both the amount of installed radiation and the gross output of the boiler by 12%, thus making no change in the piping and pick-up factor, the length of the pick-up time when using a hot water system at an outdoor temperature of −10 F was reduced from 435 to 225 min, a reduction of approximately 50%. At an outdoor temperature of 30 F the reduction in warm-up time was approximately 35%.

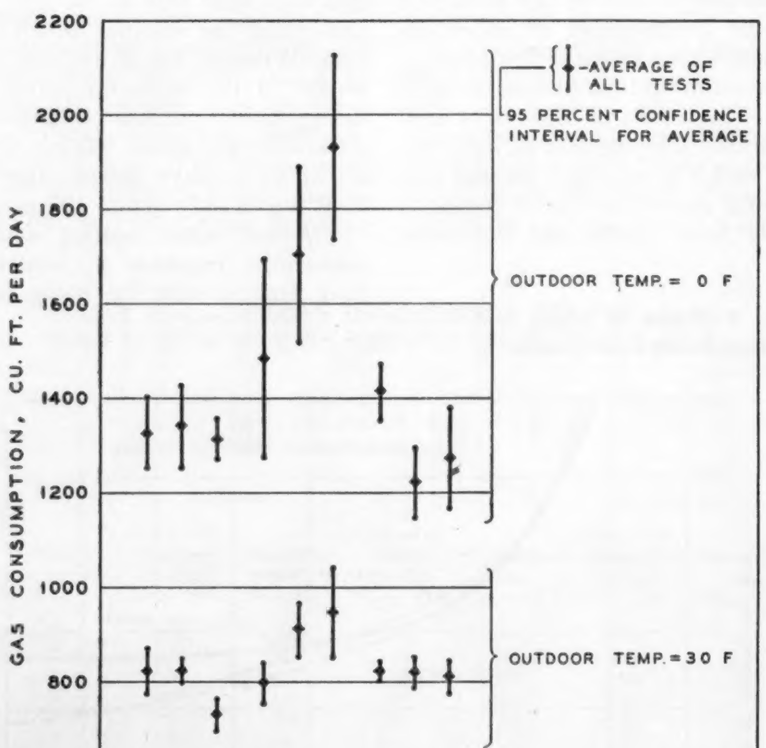
The results of these tests indicate that the warm-up time for a hot-water system may be decreased to some extent by the use of an oversized boiler, but if minimum warm-up times are desired, the entire system must be increased in size. Almost no decrease in the warm-up time results from the use of an oversized steam boiler.

DAILY FUEL CONSUMPTION

The average daily gas consumption for each series at outdoor temperatures of 0 and 30 F are shown in Fig. 4. In comparing the 1950-51 and 1956-57 tests on hot-water systems, there is an indication that the fuel consumption was slightly higher for the 1956-57 tests, although, considering the 95% confidence interval, it is doubtful that the difference is significant at an outdoor temperature of 30 F. At least a part of any increase there may have been in the daily fuel consumption in the 1956-57 tests resulted from the lower boiler efficiency observed during those tests. As an outdoor temperature of 30 F approximates average winter conditions at Urbana, Ill., the fuel consumption at this temperature is an excellent index of seasonal fuel consumption.

While the difference in daily gas consumption does appear to be significant at an outdoor temperature of 0 F, the large confidence interval indicates that additional tests at low outdoor temperatures are required to establish accurate values. During the winter of 1956-57 there were few days with average temperatures below about 15 F and as a result the fuel consumptions at 0 F for Series A-56, B-56 and C-56 shown in Fig. 4 had to be extrapolated beyond the range of actual test data. This practice in itself introduced some doubt as to the accuracy of the values.

Fig. 4 Average daily gas consumption



While there was no clear cut indication that the fuel burning rate had any effect on the daily fuel consumption in the 1950-51 tests, there was a definite increase in the daily fuel consumption as the fuel burning rate was increased in the 1956-57 tests. The range of fuel burning rates was larger in 1956-57 than in 1950-51 and at the higher burning rates the boiler was being over-fired.

Making allowance for the unfavorable winter in 1956-57 and the over-firing of the boiler in Series C-56, the fact still remains that neither oversizing the system nor increasing the piping and pick-up factor decreased the daily fuel consumption. On the contrary, they probably resulted in some increase.

The fuel consumptions obtained with the steam system in 1957-58 were approximately the same as those obtained on the hot-water system in 1950-51. However, this comparison may be misleading since the boiler used for the steam tests was designed for gas and was 10 to 15% more efficient than the all purpose boiler equipped with a conversion gas burner which was used on the hot-water system during the winters of 1950-51 and 1956-57. Had the same all purpose boiler been used on all tests, the daily fuel consumption for the steam tests would probably have been about 10 to 15% higher than those shown in Fig. 4 and would closely approximate earlier test results.[3]

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Chapter strength

—the key to society progress

PRESIDENT'S PAGE
WALTER A. GRANT
Guest Sponsor

Without question, chapter strength will be the controlling key to future progress for ASHRAE. The dynamic growth and accomplishments of both ASHAE and ASRE resulted directly from the strong leadership, sound organization and wise policies pursued by the individual chapters toward the achievement of Society objectives.

Our chapters are integral arms of the Society, created and governed by the national By-Laws,[1] and organized and operated from their own Chapter By-Laws. The latter are patterned after the Model Chapter By-Laws[2] developed by Regions Central Committee,[3] tailored to fit local situations, and subject to approval by the Charter and By-Laws Committee of the Society. The national organization encourages chapters to operate with a considerable degree of autonomy, but since in the eyes of the law they are part of the Society, their actions must always be in harmony with national policies and objectives.

To assist chapters to function in a business-like manner, Headquarters Staff in New York is currently preparing a Manual on Chapter Operations, which Regions Central expects to make available around the first of the year.

Only members of the national Society in good standing may be members of a chapter, and each such national member is eligible to join the chapter of his choice. Unless he specifies otherwise to Headquarters, he is assigned to the chapter in his own geographical territory. Chapters receive annually an allocation of funds for each national member assigned them, to help defray the cost of meeting notices sent him, and other chapter operating expenses, even though the member may not be on the active roll of the chapter.

Effective operation of a chapter is always a direct result of election by its membership of a strong slate of officers and Board of Governors, who are willing and able to devote time and effort to chapter affairs. The Chapter By-Laws provide for stand-

WALTER A. GRANT
Second Vice President, ASHRAE

ing committees to spread the load and share these responsibilities among the membership, and facilitate the development of future officer material. Of great importance to growth are the Membership, Attendance and Reception Committees, to bring new blood into the chapter, and assure a good turnout at meetings of old members, new candidates and guests.

The monthly chapter meeting is the principal vehicle for attaining the Society's objectives which are spelled out in the By-Laws. Since chapter progress is so greatly dependent on good attendance, keen interest and broad participation, the skillful planning of meetings is a matter of the deepest concern to chapter officers and governors, as well as to the Program Committee directly responsible. To avoid protracted discussions, most successful chapters transact their routine business behind the scenes at Board of Governors and committee meetings, reporting briefly to the membership on actions taken, and reserving for the chapter meeting only those business items in which the full membership must participate.

Perhaps one of the greatest challenges to chapter management today is to provide balanced programs which will interest all major segments of its membership. A few of the larger chapters can follow the pattern used successfully by the major engineering societies, which hold separate meetings for each interest section. But the majority of our chapters must necessarily rely on programs and speakers of such high quality and ability that each member may benefit substantially from exposure to subject matter unrelated to his particular business interest.

While qualified program talent is available to chapters from the Speakers' List published by Headquarters, many chapters find their most successful meetings result from giving their own members an opportunity to appear on the platform, to present technical material which may later be offered to the regional or national organization in the form of papers, or to participate in discussion panels on topics of lively and perhaps controversial interest to the chapter memberships.

Well-organized meetings of high quality not only pay off in good attendance and new members, but attract as regular guests the prominent architects, consulting engineers, educators, and contractors residing in the area. This is a two-way street, contacts, and at the same time providing an opportunity for our guests to better understand the problems of our industry.
benefiting members through valuable

Finally, the strong chapter provides an important medium through which its members may exercise progressive influence toward betterment of their community. Close liaison with industry and professional groups, such as contractors' associations and joint technical society councils, advances the stature of the chapter and improves working relationships in the area. And perhaps one of the greatest opportunities today for service to our country lies in sustained contact with the educational community—high schools, technical institutes, and engineering colleges – to encourage boys and young men to pursue technical and engineering careers. These contacts involve close relationships with teaching staffs; participation in Career Day, Engineers' Day, Science Fairs and similar events; student attendance at chapter meetings; and where feasible, formation of Student Branches of the Society.

For the chapter is indeed the grass roots strength of ASHRAE, which has written into its charter the primary purpose for its existence: to advance the arts and sciences of heating, refrigeration, air conditioning and ventilation, for the benefit of the general public.

1 ASHRAE By-Laws dated January 29, 1959
2 Model Chapter By-Laws—draft dated February 24, 1959
3 "Regional Operations" by D. D. Wile (First Vice-President) ASHRAE JOURNAL, September 1959, page 55. This article describes the relationship between the national and regional organizations and the individual chapters.

Solubility of some

Chlorofluorohydrocarbons

in tetraethylene glyco dimethyl ether

Cycle efficiency of an absorption refrigeration unit depends primarily on the heat of vaporization of the refrigerant, solubility of the refrigerant in the solvent and the vapor pressure range of the refrigerant over the operating cycle. Other practical considerations are also important for commercialization, e.g., high rate of solution; low viscosity, heat capacities and freezing points; good chemical and thermal stability; and non-corrosive, non-toxic, non-flammable, and inert components. The search for a good refrigerant-solvent combination, e.g., high rate of solution; low will, no doubt, have to consider all of the items mentioned above, in addition to other factors such as design and cost. Perhaps the most difficult requirement for absorption refrigeration is the condition of high refrigerant solubility. A trial and error search for systems which exhibit high solubility would be near impossible due to the large number of mathematical combinations. For this reason, in an effort to uncover some guiding principles, methodical investigation was made of some systems which exhibit high solubility.

S. V. R. MASTRANGELO

Zellhoeffer, Copley and Marvel[1] have determined the solubility of dichlorofluoromethane in a number of solvents ranging from simple ethers, esters, amides and amines to rather complex polymeric derivatives of these. In order to explain their data, they postulated the existence of C-H ← O or C-H ← N type of hydrogen bond formation. Since this publication, there has been much evidence[2] in favor of this hypothesis, which is rather well accepted at the present time. Much of the original solubility data of Zellhoeffer, et al.[1,3] consist of one or two points at a specified activity since they had hoped merely to observe and explain the effects of high and low solubility due to hydrogen bonding.

One of the better solvents for chlorofluorohydrocarbons which they found were the polyethylene ethers, and, especially, tetraethylene glycol dimethyl ether. It was desirable, therefore, to make a more thorough study of the solubility of various chlorofluorohydrocarbons in this solvent to observe the effect of solute on the solubility. For this purpose, the solubilities at various temperatures and pressures of Refrigerant-21 dichlorofluoromethane, Refrigerant-22 chlorodifluoromethane, chlorofluoromethane, 1,1,2,2-tetrafluoroethane and 1-chloro-1,1,2,2-tetrafluoroethane were determined in tetraethylene glycol dimethyl ether.

APPARATUS AND PROCEDURE

The solubility apparatus consists essentially of a stainless steel solvent vessel provided with several stainless steel balls for agitation, and a vessel equipped with a valve which contains the solute. A portion of the solute is transferred to the solvent vessel via the gas manifold and the equilibrium pressure read on a mercury manometer or pressure gauge. The valve to the solvent vessel is closed and the excess solute in the gas manifold is condensed back into the solute vessel by chilling to liquid nitrogen temperature. The valve on the solute vessel is closed and the vessel weighed. The loss in weight represents the amount of refrigerant dissolved in the solvent plus the small amount present in the dead space above the solution.

Fig. 1 shows a schematic diagram of the apparatus. The total volume of the solvent container (S) is 22.74 cc. [up to valve (S)]. The volume of the solute vessel

Reporting data on the solubilities of Refrigerant-21 (dichlorofluoromethane), Refrigerant-22 (chlorodifluoromethane), Refrigerant-31 (chlorofluoromethane), Refrigerant-134 (1, 1, 2, 2-tetrafluoroethane), and Refrigerant-124a (2-chlorotetrafluoroethane) measured in tetraethylene glycol dimethyl ether at a number of temperatures and pressures, this paper also reports the calculated values of the heats of mixing.

S. V. R. Mastrangelo is with the Organic Chemicals Department, E. I. duPont de Nemours and Co.
Use of the data presented in this paper for calculating cycle efficiencies and for making comparisons between systems has been developed by B. J. Eiseman, Jr., who presented "A Comparison of Fluoroalkane Absorption Refrigerants" at the ASHRAE annual meeting in Lake Placid, N. Y., June 22-24, 1959 and whose paper is planned for later publication in this JOURNAL.

(R) is 14.7 cc [up to valve (R)]. The gas manifold consists of ¼-in. copper tubing and valves which connect the solute and solvent vessels to the manometer via valve (M), the pressure gauges via valves (G) and (G′), the solute supply cylinder via valves (C) and (C′), the high vacuum line via valve (V), and to each other via valves (S), (S′), (G′), (R′) and (R).

The constant temperature bath is provided with an immersion heater (500 watt) and a regulator. Calibrated mercury thermometers were used for measuring temperature. The stirrer shaft was provided with an eccentric section which was connected to the copper tube leading to the solvent vessel. This served to agitate the solvent vessel to facilitate equilibrium. The apparatus was enclosed in a hot box for operation above room temperature.

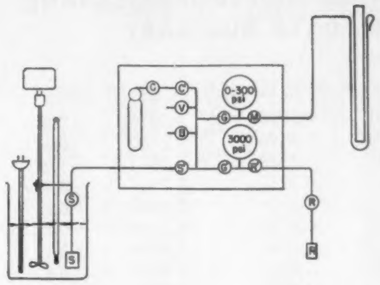
The pressure gauge was calibrated against the vapor pressure of Refrigerant-12 dichlorodifluoromethane using literature values[4] for comparison. Also, this gauge was calibrated against the manometer by expansion of air and against a dead-weight gauge. The calibration showed that this gauge was accurate to ±0.2 psi above 15 psig. However, below 15 psig, the manometer was used exclusively for the pressure measurements.

CALCULATIONS AND CORRECTIONS

The following corrections were made to the experimental data:

Dead space – The amount of dead space above the solution in (S) was calculated as the difference between the initial volume (22.74 cc) and the volume of liquid solvent plus liquid solute based on liquid density data and the weight of solvent plus solute present. Fortunately, these corrections were small (5% or less) so that knowledge of the actual solution volumes was not necessary.

Weight in dead space – The weight of solute present in the gas phase above the solution was calculated from vapor density data, when available, or from blank runs with no solvent. This correction never amounted to more than 5% of the



Schematic arrangement of the solubility apparatus

total solute so that an accuracy of 10% in this correction causes an error never greater than 0.5%.

Absolute pressure – Pressures read on the gauge were reduced to absolute pressures by measurement of the barometric pressure before and after each experimental point.

Variation in P° with bath temperature – In most cases, values of the vapor pressure of the solute were known from published data reported previously. When data were not available, the vapor pressures were measured. In some cases, an effort was made to correct the P° values for slight variations in the bath temperature in each isotherm with the assumption that P/P° varies considerably less with temperature than P alone. Values of P° above the critical pressure were extrapolated on the basis of log P° versus $\frac{1}{T}$.

Losses—After recondensation of the excess solute back into (R), an amount of gas amounting to from 0.5 to 2.0 mm was usually left in the line. This amounted to from 0.5 to about 2 mg of gas, assuming the perfect gas law, and 80 ml. for the line volume (calibrated by gas expansion from a known volume). Corrections for these losses were applied.

Material balance – At the end of each isotherm measurement, the solute was distilled back into the solute vessel and the weights obtained. The sum of the final weights of (R) + (S), minus the losses calculated by step (5), must equal the sum of the original weights of (R) and (S). In most cases, the material balance was better than 0.01 to 0.02 g out of from 6 to 15 g of solute dissolved in the final point. Whenever balances of greater than 0.02 g were obtained, they were much higher (0.2 g and higher). This indicated the presence of a leak in the system so that the measurements were repeated.

TABLE 1 REFRIGERANT-21 DICHLOROFLUOROMETHANE IN TETRAETHYLENE GLYCOL DIMETHYL ETHER (E-181) EXPERIMENTAL DATA SUMMARY

†C	X_r	P psia.	P° psia.	$P/P° \equiv a_r$	$P/P°/X_r \equiv \gamma_r$	γ_r (Calc'd)[a]	$\Delta\gamma_r$,%
27.2	0.1444	0.6613	27.03	0.0245	0.1695	0.172	1.5
27.6	0.4118	2.943	27.68	0.1063	0.2581	0.257	0.4
28.1	0.6628	8.802	28.49	0.3090	0.4662	0.432	7.9
28.5	0.7476	13.101	29.14	0.4496	0.6014	0.568	5.9
28.6	0.7815	15.392	29.30	0.5253	0.6722	0.656	2.5
28.6	0.8046	17.104	29.30	0.5838	0.7256	0.725	0.1
28.6	0.8232	18.569	29.30	0.6338	0.7699	0.799	3.6
28.6	0.8409	19.956	29.30	0.6811	0.8100	0.886	8.6
27.7	0.5949	6.261	27.84	0.2249	0.3780	0.361	4.7
28.8	0.2856	1.7075	29.62	0.0577	0.2019	0.206	2.0
						average =	±3.1
55.8	0.4793	11.361	65.90	0.1724	0.3597	0.359	0.2
56.0	0.5463	15.098	66.51	0.2270	0.4155	0.412	0.8
56.3	0.5878	17.896	67.44	0.2654	0.4515	0.449	0.6
56.3	0.6289	20.952	67.44	0.3107	0.4940	0.493	0.2
56.0	0.6430	22.011	66.51	0.3309	0.5146	0.516	0.3
						average =	±0.4
89.9	0.2776	13.557	153.9	0.0881	0.3173	0.315	0.7
89.9	0.3351	17.610	153.9	0.1144	0.3414	0.346	1.3
89.7	0.3823	21.402	152.7	0.1402	0.3667	0.371	1.2
						average =	±1.0

a—Calculated by the equation: γ_r (calc'd). $= K_1 + K_2 a_r$, where

†C	K_1	K_2
28.6	0.148	0.9905
56.0	0.190	0.983
89.7	0.226	1.030

TABLE II REFRIGERANT-22 CHLORODIFLUOROMETHANE IN E-181 EXPERIMENTAL DATA SUMMARY

t C	X_r	P psia.	P° psia.	$P/P^\circ \equiv a_r$	$P/P^\circ/X_r \equiv \gamma_r$	γ_r (Calc'd)[a]	$\Delta\gamma_r$,%
28.65	0.1553	5.074	170.10	0.02983	0.1921	0.1931	0.5
28.4	0.2548	9.204	168.25	0.05470	0.2146	0.2160	0.6
28.9	0.3720	15.821	171.95	0.09201	0.2473	0.2503	1.2
28.5	0.4579	22.879	168.99	0.1354	0.2957	0.2901	1.9
28.4	0.4981	25.173	168.25	0.1497	0.3004	0.3032	0.9
28.8	0.5269	28.595	171.21	0.1670	0.3170	0.3192	0.7
28.4	0.5775	34.4	168.25	0.2045	0.3541	0.3536	0.1
28.2	0.6226	44.2	166.77	0.2470	0.3967	0.3927	0.9
28.4	0.6763	50.0	168.25	0.2972	0.4394	0.4388	0.1
28.6	0.7414	67.7	169.73	0.3989	0.5380	0.5323	1.1
29.05	0.8145	89.5	173.07	0.5171	0.6349	0.6409	0.9
						average =	±0.8
56.0	0.08157	6.149	326.1	0.01886	0.2312	0.2176	6.3
56.0	0.1623	12.960	326.1	0.03974	0.2449	0.2364	3.6
56.0	0.1052	7.911	326.1	0.02426	0.2306	0.2225	3.6
55.7	0.1736	12.335	323.9	0.03808	0.2194	0.2349	6.5
55.8	0.2245	17.457	324.1	0.05386	0.2399	0.2492	3.7
56.0	0.2642	22.118	326.1	0.06782	0.2567	0.2618	5.8
56.0	0.3525	33.6	326.1	0.1030	0.2921	0.2936	5.1
56.0	0.4613	50.6	326.1	0.1552	0.3364	0.3408	1.3
56.0	0.5607	74.8	326.1	0.2294	0.4091	0.4079	0.3
56.1	0.6021	87.1	326.8	0.2665	0.4426	0.4415	0.2
56.2	0.6391	100.2	327.5	0.3060	0.4788	0.4772	0.3
56.1	0.6468	102.6	326.8	0.3140	0.4855	0.4844	0.2
						average =	±3.1
85.8	0.08162	13.182	606.8	0.02172	0.2661	0.2624	1.4
86.0	0.1182	19.374	609.1	0.03181	0.2691	0.2712	0.8
85.9	0.1626	27.814	607.9	0.04576	0.2814	0.2834	0.7
85.7	0.2283	41.1	605.6	0.06787	0.2973	0.3027	1.8
86.0	0.3019	58.7	609.1	0.09637	0.3192	0.3276	2.6
86.1	0.3288	73.5	610.3	0.1204	Discarded point		
86.0	0.3952	91.5	609.1	0.1502	Discarded point		
85.7	0.4260	101.1	605.6	0.1669	0.3918	0.3892	0.7
86.0	0.4743	113.7	609.1	0.1867	0.3936	0.4065	3.2
86.3	0.1118	18.904	612.6	0.03086	0.2760	0.2704	2.1
86.2	0.1979	31.1	611.4	0.05087	Discarded point		
86.2	0.2988	52.1	611.4	0.08521	Discarded point		
86.0	0.3499	67.3	609.1	0.1105	Discarded point		
86.0	0.4129	85.4	609.1	0.1402	Discarded point		
86.8	0.2771	55.2	618.5	0.08925	0.3221	0.3214	0.2
86.3	0.3336	72.0	612.6	0.1175	0.3522	0.3460	1.8
86.4	0.4014	92.0	613.6	0.1499	0.3734	0.3743	0.2
						average =	±1.4
91.3	0.07600	13.98	673.5	0.02076	0.2732	0.2702	1.1
91.4	0.1020	19.13	674.8	0.02835	0.2779	0.2765	0.5
90.8	0.1525	29.6	667.3	0.04436	0.2909	0.2896	0.4
90.8	0.2118	43.1	667.3	0.06459	0.3049	0.3062	0.4
90.9	0.2994	61.2	668.5	0.09155	0.3058	0.3283	(6.9)
90.7	0.3383	80.5	666.0	0.1209	0.3574	0.3524	1.4
90.6	0.3926	98.1	664.8	0.1477	0.3762	0.3743	0.5
90.8	0.4174	107.2	667.3	0.1606	0.3848	0.3849	0.0
91.2	0.4408	114.6	672.2	0.1705	0.3868	0.3930	1.6
						average =	±0.7
111.1	0.4690	190.0	910[b]	0.2088	0.4452	0.4450	0.05
111.1	0.4964	208.4	910	0.2290	0.4613	0.4612	0.03
111.1	0.6201	311.5	910	0.3423	0.5520	0.5520	0.00
						average =	±0.03
148.9	0.2903	171.8	1130[b]	0.1520	0.5236	0.5233	0.06
148.9	0.3223	197.9	1130	0.1751	0.5433	0.5417	0.29
148.9	0.4414	310.4	1130	0.2747	0.6224	0.6233	0.14
						average =	±0.18
176.7	0.2193	174.8	1238[b]	0.1412	0.6439	0.6453	0.24
176.7	0.2454	200.7	1238	0.1621	0.6606	0.6609	0.05
176.7	0.3448	310.3	1238	0.2506	0.7268	0.7270	0.03
						average =	±0.11

a—Calculated by the equation: γ_r (calc'd). $= K_1 + K_2 a_r$, where

t C	K_1	K_2
28.6	0.1657	0.9191
56.0	0.2005	0.9042
86.0	0.2434	0.8735
91.0	0.2532	0.8202
111.1	0.2776	0.8017
148.9	0.4000	0.8117
176.7	0.5399	0.7468

eb—Based on a large extrapolation above the critical temperature.

Thermodynamical parameters – The parameters used in the later discussion, as calculated from the experimental data, are defined below

$$\text{Mole fraction of solute} = X_r \equiv \frac{W_r/M_r}{W_r/M_r + W_s/M_s}$$

where W_r and W_s are the weights, and M_r and M_s the molecular weights of solute and solvent, respectively. The mole fraction of solvent, X_s, is simply $(1\text{-}X_r)$.

$$\text{Activity of the solute} = a_r \equiv P/P^\circ$$

where P is the partial pressure of solute vapor above the solution, and P° is the saturated vapor pressure of the solute at the same temperature.

$$\text{Activity coefficient of the solute} = \gamma_r \equiv a_r/X_r$$

Materials used as solutes were all obtained as middle-cuts in precise distillations. Infrared and mass spectroscopic techniques were used for identification and quality evaluation. Also, gas chromatography was used to determine impurities where identification was difficult with other spectroscopic techniques. The purity of all of the materials used was 99.5% or better. The tetraethylene glycol dimethyl ether was "Ansul Ether E-181." This material was used without further purification.

RESULTS AND DISCUSSION

The solubilities at various temperatures and pressures of Refrigerant-21 dichlorofluoromethane, Refrigerant-22 chlorodifluoromethane, chlorofluoromethane, 1,1,2,2-tetrafluoroethane and 1-chloro-1,1,2,2-tetrafluoroethane in tetraethylene glycol dimethyl ether are listed in Tables I-IV, respectively. These data were fitted empirically to an equation of the form shown by equation (1):

$$\gamma_r = K_1 + K_2 a_r \tag{1}$$

where γ_r is the activity coefficient of the solute, a_r is its activity, and K_1 and K_2 are constants. The precision of this fit is good except at quite low values of the mole fraction (where large weighing errors may occur) and at the quite high temperatures where dead space

corrections become important in view of the high pressures and low solubilities.

The data were interpolated to rounded values of the mole fraction and plotted as log γ_r versus $1/T°K$. The slope of the straight line obtained was used for calculating the partial molal heats of dilution of the solutes according to equation (2).

$$\Delta\overline{H}_r = 2.303\ R \left[\frac{\Delta \log \gamma_r}{\Delta \log \frac{1}{T}} \right]_{X_r} \qquad (2)$$

These heats are listed in Table VI. The accuracy of these values is approximately $\pm 10.0\%$.

Comparison of the relative solubilities of the chlorofluorohydrocarbons studied is rather difficult because of the relative variation not only of the intercepts [K_1 of equation (1)], but also of the slopes [K_2 of equation (1)]. Comparison at 56 C of the intercepts of equation (1) indicates that $CHClF_2 > CHCl_2F >> CHF_2CHF_2 > CH_2ClF >> CHF_2CClF_2$. However, the slope of the CHF_2CClF_2 line is only about 2/3 of the slopes of the CHF_2CHF_2 and CH_2ClF lines. Hence, in regions of moderate solubility ($a_r > 0.3$), the solubility of these three materials is about the same. The only obvious difference in solubility exists between the trihalomethanes, and the dihalomethane and haloethanes. The relatively low solubility of chlorofluoromethane and 1,1,2,2-tetrafluoroethane is believed to result from the presence of two hydrogens per molecule which can accept electrons from two ether sites in the solvent. Steric factors are also partially involved in the cases of 1,1,2,2-tetrafluoroethane and 1-chloro-1,1,2,2-tetrafluoroethane. A further discussion of these results is presented in another paper.[5]

TABLE III CHLOROFLUOROMETHANE IN E-181

t C	X_r	P psia.	P° psia.	$P/P° \equiv a_r$	$P/P°/X_r \equiv \gamma_r$	γ_r (Calc'd)[a]	$\Delta\gamma_r$%
35.0	0.5908	22.33	71.2	0.3136	0.5308	0.5297	0.21
35.0	0.6258	25.29	71.2	0.3552	0.5676	0.5681	0.09
35.0	0.6485	27.46	71.2	0.3857	0.5948	0.5964	0.27
35.0	0.6815	31.60	71.2	0.4438	0.6512	0.6501	0.17
						average =	±0.19
54.4	0.4341	23.25	123.5	0.1883	0.4338	0.4341	0.07
54.4	0.4612	25.84	123.5	0.2092	0.4536	0.4534	0.04
54.4	0.4866	28.34	123.5	0.2295	0.4716	0.4718	0.04
54.4	0.5238	32.70	123.5	0.2648	0.5055	0.5048	0.14
						average =	±0.07
73.9	0.2996	23.33	202.5	0.1152	0.3845	0.3825	0.52
73.9	0.3230	25.78	202.5	0.1273	0.3941	0.3937	0.10
73.9	0.2997	23.13	202.5	0.1142	0.3810	0.3815	0.13
73.9	0.3228	25.69	202.5	0.1296	0.3931	0.3958	0.68
73.9	0.3475	28.48	202.5	0.1406	0.4046	0.4060	0.34
73.9	0.3794	32.95	202.5	0.1627	0.4288	0.4264	0.56
						average =	±0.39
111.1	0.3283	61.11	45.6	0.1340	0.4082	0.4090	0.20
111.1	0.3698	71.90	45.6	0.1577	0.4264	0.4274	0.23
111.1	0.5217	122.1	45.6	0.2677	0.5131	0.5127	0.08
111.1	0.2154	36.03	45.6	0.07901	0.3668	0.3663	0.14
						average =	±0.16
148.9	0.1930	60.85	895	0.06799	0.3523	0.3527	0.12
148.9	0.1440	44.70	895	0.04994	0.3468	0.3401	(1.97)
148.9	0.1899	60.40	895	0.06749	0.3554	0.3524	0.85
148.9	0.2224	71.98	895	0.08042	0.3616	0.3615	0.03
148.9	0.3569	130.05	895	0.1453	0.4071	0.4070	0.03
148.9	0.3415	122.80	895	0.1372	0.4017	0.4013	0.10
148.9	0.1065	33.50	895	0.03743	0.3515	0.3313	0.06
148.9	0.1457	45.3	895	0.0506	0.3473	0.3405	(1.99)
148.9	0.2141	68.6	895	0.07665	0.3580	0.3588	0.22
148.9	0.2800	94.4	895	0.1055	0.3768	0.3791	0.61
148.9	0.3513	126.8	895	0.1416	0.4031	0.4044	0.32
						average =	±0.26
176.7	0.1365	61.8	1070	0.05776	0.4232	0.4225	0.17
176.7	0.1590	72.67	1070	0.06792	0.4272	0.4282	0.23
176.7	0.2022	94.73	1070	0.08853	0.4378	0.4398	0.45
176.7	0.2554	124.4	1070	0.1163	0.4554	0.4554	0.00
						average =	±0.21

a—Calculated by the equation: γ_r (calc'd). $= K_1 + K_2a_r$, where

t C	K_1	K_2	t°C.	K_1	K_2
35.0	0.2396	0.9250	111.1	0.3050	0.7760
54.4	0.2599	0.9250	148.9	0.3050	0.7020
73.9	0.2759	0.9250	176.7	0.3900	0.5620

TABLE IV 1,1,2,2-TETRAFLUOROETHANE IN E-181

t C	X_r	P psia.	P° psia.	$P/P° \equiv a_r$	$P/P°/X_r \equiv \gamma_r$	γ_r (Calc'd)[a]	$\Delta\gamma_r$%
28.8	0.1875	4.206	86.29	0.04874	0.2599	0.2644	1.70
28.2	0.2942	7.710	84.72	0.09101	0.3093	0.3092	0.03
28.6	0.3624	10.98	85.77	0.1280	0.3532	0.3483	1.41
28.2	0.4102	13.40	84.72	0.1582	0.3857	0.3802	1.44
28.6	0.4422	15.66	85.77	0.1826	0.4111	0.4059	1.28
28.6	0.4647	17.22	85.77	0.2008	0.4321	0.4252	1.62
28.6	0.5622	27.69	85.77	0.3228	0.5701	0.5541	2.89
28.7	0.6235	32.2	86.03	0.3743	0.6003	0.6085	1.36
28.2	0.6758	40.1	84.72	0.4733	0.7004	0.7131	1.78
28.6	0.7398	48.1	85.77	0.5608	0.7580	0.8056	(5.91)
						average =	±1.5
56.7	0.2260	15.22	187.7	0.08109	0.3588	0.3595	0.19
56.3	0.3000	22.26	185.8	0.1198	0.3987	0.3951	0.91
56.0	0.3824	31.2	184.4	0.1692	0.4425	0.4410	0.34
56.0	0.4769	45.0	184.4	0.2440	0.5116	0.5102	0.27
56.0	0.5653	61.6	184.4	0.3341	0.5910	0.5935	0.42
56.0	0.5906	67.3	184.4	0.3650	0.6180	0.6221	0.66
						average =	±0.47
86.4	0.1305	19.08	348[b]	0.05483	0.4202	0.4085	(2.86)
86.2	0.2076	31.6	348	0.09080	0.4374	0.4377	0.07
86.3	0.2984	49.9	348	0.1434	0.4806	0.4804	0.04
86.2	0.3685	66.2	348	0.1902	0.5161	0.5183	0.42
86.2	0.3941	73.4	348	0.2109	0.5351	0.5351	0.00
						average =	±0.13

a—Calculated by the equation: γ_r (calc'd). $= K_1 + K_2a_r$, where

t C	K_1	K_2
26.6	0.2130	1.05667
56.0	0.2845	0.92500
86.0	0.3640	0.81143

b—Based on a large extrapolation from room temperature.

REFERENCES

1. G. F. Zellhoeffer, M. J. Copley and C. S. Marvel, J Am. Chem. Soc. *60*, 1337 (1938).

2. For example, see:
a. G. M. Barrow and E. A. Yerger, J. Am. Chem. Soc. *76*, 5211 (1954); *77*, 4474 (1955), *77*, 6206 (1955); and G. M. Barrow, J. Am. Chem. Soc. *78*, 5802 (1956).
b. E R. Lippincott and R. Shroeder, J. Chem. Phys. *23*, 1099 (1955).
c. S. Nagakura and M. Gouterman, J. Chem. Phys. *26*, 881 (1957).

TABLE V 1-CHLORO-1,1,2,2-TETRAFLUOROETHANE IN E-181 EXPERIMENTAL DATA SUMMARY

t C	X_r	P psia.	P° psia.	$P/P° = a_r$	$P/P°/X_r \equiv \gamma_r$	γ_r (Calc'd)[a]	$\Delta\gamma_r$,%
35.0	0.5576	24.53	70.60	0.3475	0.6232	0.6109	2.01
35.0	0.5923	26.27	70.60	0.3721	0.6282	0.6278	0.06
35.0	0.6279	29.04	70.60	0.4114	0.6558	0.6546	0.18
35.0	0.7398	32.45	70.60	0.4597	0.6213	0.6875	—
35.0	0.5526	23.32	70.60	0.3303	0.5977	0.5993	0.27
35.0	0.5963	26.05	70.60	0.3690	0.6118	0.6257	2.22
35.0	0.6274	28.91	70.60	0.4095	0.6527	0.6533	0.09
35.0	0.6635	32.65	70.60	0.4625	0.6971	0.6894	1.12
35.0	0.5540	23.36	70.60	0.3309	0.5973	0.5997	0.40
35.0	0.5928	26.22	70.60	0.3714	0.6265	0.6273	0.13
35.0	0.6281	29.03	70.60	0.4054	0.6454	0.6505	0.78
35.0	0.6644	32.70	70.60	0.4632	0.6972	0.6899	1.06
35.0	0.7323	39.65	70.60	0.5617	0.7670	0.7571	1.31
35.0	0.7718	44.12	70.60	0.6250	0.8098	0.8003	1.19
35.0	0.2341	7.695	70.60	0.1090	0.4656	0.4483	3.86
35.0	0.3501	12.264	70.60	0.1737	0.4962	0.4925	0.75
35.0	0.4477	17.02	70.60	0.2410	0.5383	0.5384	0.02
						average =	±1.0
54.4	0.3461	23.35	119.7	0.1950	0.5634	0.5598	0.64
54.4	0.3836	26.25	119.7	0.2193	0.5717	0.5743	0.45
54.4	0.4197	29.48	110.7	0.2463	0.5868	0.5903	0.59
54.4	0.4416	32.55	119.7	0.2719	0.6157	0.6055	1.66
54.4	0.3636	24.66	119.7	0.2060	0.5666	0.5664	0.04
54.4	0.3836	26.28	119.7	0.2195	0.5722	0.5744	0.38
54.5	0.4199	29.53	119.87	0.2464	0.5868	0.5904	0.61
54.5	0.4535	32.75	119.87	0.2732	0.6024	0.6063	0.64
54.4	0.1608	9.852	119.7	0.08223	0.5119	0.4929	3.85
54.4	0.5381	42.15	119.7	0.3521	0.6543	0.6531	0.18
54.4	0.6317	54.10	119.7	0.4520	0.7155	0.7125	0.42
54.4	0.7029	64.35	119.7	0.5376	0.7648	0.7633	0.20
54.4	0.1625	9.909	119.7	0.08278	0.5095	0.4932	3.19
54.4	0.2674	17.067	119.7	0.1426	0.5332	0.5287	0.85
54.4	0.5397	42.45	119.7	0.3546	0.6570	0.6546	0.37
54.4	0.6349	54.20	119.7	0.4528	0.7132	0.7070	0.88
54.4	0.7127	65.65	119.7	0.5485	0.7696	0.7698	0.03
						average =	±0.9
73.9	0.2184	23.764	190.6	0.1247	0.5710	0.5683	0.48
73.9	0.2413	26.457	190.6	0.1388	0.5752	0.5755	0.05
73.9	0.2652	29.415	190.6	0.1543	0.5818	0.5834	0.26
73.9	0.2882	32.60	190.6	0.1710	0.5927	0.5919	0.14
73.9	0.4651	58.20	190.6	0.3054	0.6566	0.6601	0.53
73.9	0.5658	76.13	190.6	0.3994	0.7059	0.7079	0.28
73.9	0.6549	94.83	190.6	0.4975	0.7598	0.7577	0.28
						average =	±0.3
111.1	0.1703	42.15	407	0.1036	0.6083	0.5994	1.48
111.1	0.2391	60.2	407	0.1479	0.6186	0.6197	0.17
111.1	0.2816	72.7	407	0.1786	0.6342	0.6338	0.07
111.1	0.3543	94.7	407	0.2327	0.6568	0.6586	0.27
111.1	0.4352	121.0	407	0.2973	0.6831	0.6882	0.74
						average =	±0.55
148.9	0.1279	60.55	774[b]	0.07823	0.6116	0.6167	0.82
148.9	0.1564	74.30	774	0.09599	0.6137	0.6235	1.57
148.9	0.1960	93.73	774	0.1211	0.6179	0.6330	2.39
148.9	0.2308	117.4	774	0.1517	0.6573	0.6446	1.97
148.9	0.2379	121.9	774	0.1575	0.6620	0.6469	2.33
						average =	±1.82
176.8	0.09155	61.1	1035[b]	0.05903	0.6448	0.6433	0.23
176.8	0.1095	73.25	1935	0.07077	0.6463	0.6468	0.08
176.8	0.1419	94.7	1035	0.09150	0.6448	0.6529	1.24
176.85	0.1777	119.2	1035	0.1152	0.6483	0.6599	1.76
176.7	0.09122	62.5	1035	0.06039	0.6620	0.6438	2.83
176.7	0.1078	73.4	1935	0.07092	0.6578	0.6468	1.70
176.7	0.1391	94.5	1035	0.09130	0.6564	0.6528	0.55
176.7	0.1732	118.05	1035	0.1141	0.6588	0.6595	0.11
						average =	±1.06

a—Calculated by the equation: γ_r (calc'd). $= K_1 + K_2 a_r$, where

t C	K_1	K_2
35.0	0.374	0.682
54.4	0.444	0.594
73.9	0.505	0.508
111.1	0.552	0.458
148.9	0.587	0.380
176.8	0.626	0.294

b—Extrapolated beyond the critical pressure on the basis of log P° versus $\frac{1}{T}$.

TABLE VI PARTIAL MOLAL HEATS OF DILUTION PARTIAL MOLAL HEAT OF DILUTION — KCAL./MOLE

X_r	$CHCl_2F$	$CHClF_2$	CH_2ClF	CHF_2CHF_2	CHF_2CClF_2
0.1	—	1.43	0.89	1.91	1.55
0.2	1.58	1.40	0.83	1.78	1.41
0.3	1.49	1.38	0.74	1.63	1.31
0.4	1.47	1.34	0.65	1.39	1.15
0.5	1.45	1.29	0.53	1.10	0.93
0.6	1.32	1.23	0.37	0.78	0.61
0.7	1.14	1.08	—	—	0.39
0.8	—	—	—	—	0.24

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e. H. H. Jaffe, J. Am. Chem. Soc. 79, 3323 (1957).

f. A. M. Benson, Jr. and H. G. Drickamer, Disc. Far. Soc. 39 (1957); J. Chem. Phys. 27, 1164 (1957).

3. G. F. Zellhoeffer and M. J. Copley, J. Am. Chem. Soc. 60, 1343 (1938).

4. R. C. McHarness, B. J. Eiseman, Jr. and J. J. Martin, Refrig. Eng. 63, 31 (1955).

5. S. V. R. Mastrangelo, J. Phys. Chem. 63, 608 (1959).

WHO'S WHO IN ASHRAE

At intervals there appear in the ASHRAE JOURNAL several listings of officers, committeemen and Region and Chapter personnel.

As it is not feasible to repeat these listings each month, though changes do occur in each with some frequency, the accompanying summarizing reference may be anticipated in adequate revision in each future issue of the JOURNAL.

Insofar as possible, the listings will each appear twice a year.

ASHRAE OFFICERS, COMMITTEES

See page 76, September JOURNAL

REGION AND CHAPTER OFFICERS

See page 78, this issue

RESEARCH AND TECHNICAL COMMITTEES

See page 84, July JOURNAL

STANDARDS COMMITTEE

See page 75, June JOURNAL

TECHNICAL COMMITTEES

See page 96, May JOURNAL

INTER-SOCIETY COMMITTEES

See page 80, July JOURNAL

National conference

A. T. BOGGS, III
ASHRAE Technical Secretary

A highlight of the **Tenth National Conference on Standards,** to be held in Detroit October 20-22, will be an address by Major General J. B. Medaris, Commanding General of the Army Ordnance Missile Command. General Medaris will speak on "Standards for Survival" at the Awards Dinner on October 21 when the Howard Coonley and Standards Medals will be presented. These two medals are recognized as being the country's highest honors in the field of standards work. This annual event of ASA will have 8 technical sessions covering such subjects as space, nuclear energy, metrology, and the development and use of standards. The National Conference is a forum through which industry, government, and the public can keep abreast of the latest developments in standardization.

ASHRAE Proposed Standard 30P on the testing of liquid chilling packages appears in this issue beginning on page 70.

The Commonwealth of Massachusetts has given due recognition to the efforts of standards engineers through the proclamation by Governor Furcolo in which the week of September 20-26, 1959, was proclaimed as **Standards Engineers Week.**

DEPARTMENT OF COMMERCE: A new list of Commercial Standards revised through July 1, 1959, is now available from the U. S. Dept. of Commerce. The new index lists all Commercial Standards in 22 classifications. One of these categories is heating, ventilating, and refrigeration. New standards in preparation and existing standards being revised are shown in a supplemental list. Single copies are available without charge upon request to the Commodity Standards Div, U. S. Dept. of Commerce, Washington 25, D.C. Ask for Catalog No. 978.

CS195-57: A proposed revision to Commercial Standard CS195-57 designated TS-5453, Warm Air Furnaces Equipped with Pressure-Atomizing or Rotary-Type Oil Burners has been distributed to industry for comment.

ASA: The National Fire Protection Association has submitted for approval as an American Standard the proposed Standard for Installation of Blower and Exhaust Systems for Dust, Stock and Vapor Removal or Conveying. The Standard is a revision to ASA Z33.1-1950 and is identical with the earlier edition except for the omission of one section concerned with the ventilation of restaurant-type cooking equipment. The ventilation of restaurant-type cooking equipment has been assigned to the NFPA Committee on Chimneys and Heating Equipment. The proposed revision is being considered by the Safety Standards Board of ASA.

ARI: The first ARI standard for refrigeration units designed for use in **refrigerating trucks and trailers** was published in September. The new publication is numbered ARI Standard 1110-59 and is titled "ARI Standard for Speed-Governed Transport Refrigeration Units Employing Forced-Circulation Air Coolers." A speed-governed unit is defined as one whose drive is designed to operate at a constant, although possibly adjustable, speed.

It has been announced that a second equipment standard covering hydraulic-drive and other types of variable-speed units is being developed. Work has been initiated on an application standard to relate standard ratings and other performance characteristics of both types of units to a certification program. In its statement of purpose the standard indicates that it is published "to provide a specification of what constitutes minimum standard equipment, rating requirements, methods of testing, and proper marking for speed-governed transport refrigeration units employing forced-circulation air coolers."

Both high temperature cooling units and low temperature units are covered in the standard which establishes standard rating conditions for the testing and rating of the units and provides for the publication of ratings including a statement of cooling capacity in terms of Btuh. Copies are available from ARI at 75c each.

AFI: A recent release indicates that the **Air Filter Institute** has standardized electronic air cleaner terminology and has established certified ratings for these devices. The testing method specified by AFI is the National Bureau of Standards Dust Spot method. This method uses atmospheric air without the addition of other dust or contaminants. Other specifications approved by AFI state that: (1) capacity tables be based on efficiencies of 85 per cent, 90 per cent, and 95 per cent expressed in cfm and rounded out to the nearest tenth, (2) no claims will be made for collecting particles smaller than 0.01 micron, (3) an AFI certification of ratings be based on a well-defined testing procedure established by a test code committee and using the NBS dust spot method, and (4) a Standards Committee be established to supervise tests, approve laboratories for certified testing, and make recommendations on the issuance of certification seals.

Proposed Standard

METHODS OF TESTING FOR RATING LIQUID CHILLING PACKAGES

This proposed standard was developed by Project Committee 30P under jurisdiction of the ASHRAE Standards Committee. The final draft was approved by the project committee January 29, 1959. After review the proposed standard was approved by the Standards Committee April 2, 1959. The Board of Directors approved the proposed standard June 10, 1959 and are recommending it to the voting membership. Vote will be taken at the General Session of the Semiannual Meeting February 1, 1960.

Comments are solicited and should be sent to the Technical Secretary, ASHRAE Headquarters, before December 31, 1959.

SECTION 1. PURPOSE AND SCOPE

1.1 Purpose

1.1.1 This standard prescribes methods of testing for rating of liquid chilling packages.

1.2 Scope

1.2.1 This standard includes the types of liquid chilling packages described in Section 2 "Classifications".

1.2.2 This standard excludes the self-contained mechanically refrigerated drinking water coolers covered in ASRE Standard 18-56, and bottled beverage coolers covered in ASRE Standard 32-57.

1.2.3 This standard excludes liquid chilling packages which do not include all of the following three major components: a reciprocating compressor, a condenser, and a liquid cooler.

1.2.4 The testing provisions of this standard do not apply to packaged chillers with remotely located components with refrigerant piping provided in the field, unless the manufacturer completely defines the piping as to length, diameter, elevation, and the installation is in accordance with this definition

SECTION 2. CLASSIFICATIONS

2.1 Types

2.1.1. Water Cooled Liquid Chiller package consisting of one or more; reciprocating compressors, liquid coolers, and water cooled condensers, with all necessary accessories and controls for operation of the package. Package is factory designed and prefabricated (may be shipped as one or more components).

2.1.2 Air Cooled Liquid Chilling package consisting of one or more; reciprocating compressors, liquid coolers, and air cooled condensers, with all necessary accessories and controls for operation of the package. Package is factory designed and prefabricated (may be shipped as one or more components).

2.1.3 Evaporatively Cooled Liquid Chilling package consisting of one or more; reciprocating compressors, liquid coolers, and evaporatively cooled condensers, with all necessary accessories and controls for operation of the package. Package is factory designed and prefabricated (may be shipped as one or more components).

SECTION 3. RATING TERMS

3.1 *Ratings of Liquid Chilling Packages* may be expressed in the following terms:

3.1.1 General

a. Net refrigeration effect, Btuh or tons.
b. Temperature of liquid leaving cooler, deg F.
c. Liquid flow rate through the cooler, gpm or lb per hr.
d. Description of liquid sufficient to define its physical properties.
e. Liquid pressure drop through the cooler, psi or feet of liquid flowing.
f. Power input to compressor at rating conditions, bhp or Kw.
g. Power input to auxiliaries included as part of the package, at rating conditions, bhp or Kw.

3.2 *Ratings for Units with Water-cooled Condensers* may include terms under "General" and in addition the following:

a. Temperature of water entering or leaving condenser, deg F.
b. Water flow rate, gpm or lb per hr.
c. Water pressure drop through the condenser, psi or ft of water.

3.3 *Ratings for Units with Evaporatively-cooled Condensers* may include terms under "General" and in addition the following:

a. Dry bulb and wet bulb temperature of the condenser air entering the unit.
b. The external resistance, in. of water gauge, and the air flow, cfm of standard air.
c. Water supply, gpm or lb per hr.

3.4 *Ratings for Units with Air-cooled Condensers* may include terms under "General" and in addition the following:

a. Dry bulb temperature of condenser air entering unit, deg F.
b. The external resistance, in. of water gauge, and the air flow, cfm of standard air.

SECTION 4. TEST METHODS

4.1 Test Method

4.1.1 The test will determine net cooling capacity on the liquid side of the cooler.

4.1.2 After steady state conditions have been established, the required readings shall be taken every twenty minutes and the test shall be continued until four consecutive sets of readings are within the specified limits.

4.1.3 The test shall be applicable to all types of liquid chilling units listed in Section 2, "Classifications".

4.1.4 The test shall consist of a measurement of the net heat removed from the liquid in Btuh or tons as it

passes through the chiller by determination of the following:

a. Description of liquid sufficient to obtain necessary physical properties.
b. Liquid flow rate.
c. Enthalpy difference between entering and leaving liquid.

4.1.5 The heat removed from the liquid is the product of the liquid flow rate and the enthalpy difference.

4.1.6 The test shall also include the simultaneous determination of the compressor power required to produce the refrigeration effect. This power shall be determined by measurement of watts input to motor drive or compressor bhp.

4.2 Substantiating Tests

4.2.1 Substantiating tests are not required by this standard.

4.2.2 A substantiating test may be conducted simultaneously with the standard test. The substantiating test may use capacity determination by means of one or more of the test methods described in ASRE Standard 14-59, Methods of Testing for Rating Mechanical Condensing Units.

4.3 Air Quantity Tests

4.3.1 For air-cooled or evaporatively-cooled units, air quantities and external resistances may be determined independently of the standard rating test by methods in accordance with ASRE Standard 20. (ASRE Std. Methods of Rating and Testing Evaporative Condensers).

SECTION 5. INSTRUMENTS

5.1 Instruments

5.1.1 Instruments shall be selected from the types listed in "ASRE Standard 14-59—Methods of Testing for Rating Mechanical Condensing Units".

5.1.2 Accuracy of instruments selected shall be in accordance with that specified in "ASRE Standard 14-59—Methods of Testing for Rating Mechanical Condensing Units".

5.1.3 If flowmeters are used, the flowmeters must be constructed and installed in accordance with the applicable portion of the latest ASME Power Test Code.

5.1.4 Duplicate instrumentation shall be used for measurements 6.1.1 a, b, and c.

5.2 Tables of Physical Properties

5.2.1 Physical data used in this test procedure shall be obtained from International Critical Tables or the current issue of the ASRE Air Conditioning Refrigerating Data Book, Design Volume.

SECTION 6. MEASUREMENTS

6.1 Data to be recorded after steady state conditions have been established.

6.1.1 General Test Data

a. Temperature of entering liquid, deg F. (See 5.1.4)
b. Temperature of leaving liquid, deg F (See 5.1.4)
c. Liquid flow rate, gpm or lb. per hr. (See 5.1.4)
d. Description of liquid sufficient to obtain necessary physical properties.
e. Temperature of cooler ambient air, deg F.
f. Power input to compressor, bhp or Kw.

6.1.2 Additional data for packaged chillers with water cooled condensers.

a. Temperature of entering condenser water, deg F.
b. Temperature of leaving condenser water, deg F.
c. Condenser water flow rate, gpm or lb. per hr.

6.1.3 Additional data for packaged chillers with air cooled or evaporatively-cooled condensers.

a. Barometric pressure, in. Hg
b. Dry bulb temperature of air entering condenser, deg F
c. Wet bulb temperature of air entering condenser, deg F
d. Power input to fans and pumps, bhp or Kw
e. Water supply quantity to evaporative condenser, gpm or lb. per hour
f. Water supply temperature to evaporative condenser, deg. F
g. Fan RPM
h. External air resistance, inches of water

6.2 Auxiliary data to be recorded for general information.

6.2.1 Chilled liquid pressure drop through the cooler, psi or feet of liquid flowing.

6.2.2 Condenser water pressure drop (for water-cooled condensers), psi or feet of water.

6.2.3 Sufficient description to enable duplication of the liquid chilling package.

6.2.4 Packaged chiller nameplate data including make, model, size.

6.2.5 All motor nameplate data.

6.2.6 Specified refrigerant.

6.2.7 Operating refrigerant charge, lbs.

6.2.8 Data, place and time of test.

6.2.9 Test operator's name.

SECTION 7. TEST PROCEDURE

7.1 Preparation

7.1.1 The liquid chilling package which has been leak tested, dehydrated, evacuated and charged with the operating amount of refrigerant, shall be connected with the necessary instruments and auxiliary equipment.

7.1.2 The liquid chilling package shall be started and operated for a total accumulated time of not less than 24 hr prior to rating tests.

7.2 Operation and limits

7.2.1 Start the system and obtain and maintain the conditions in accordance with the following tolerances and instructions:

a. The individual readings of all liquid temperatures shall not vary from the specified values by more than one-half (0.5) deg F. Care must be taken to insure that these liquid temperatures are the average bulk stream temperatures.
b. The arithmetic average of all required dry bulb air temperature readings shall not vary from the specified values by more than one (1) deg F, nor shall the individual readings vary by more than this amount from the average value.
c. The arithmetic average of all required wet bulb air temperature readings shall not vary from the specified values by more than one-half (0.5) deg F nor shall the individual readings vary by more than this amount from the average value.
d. The liquid flow rate as determined by the two methods of measurement shall agree such that the difference of the two values shall not be greater than 2% of the greater value. The arithmetic mean value shall be used in the calculations of Paragraph 7.3.1.

7.2.2 After establishment of steady state conditions, all required readings shall be taken every twenty minutes and the test shall be continued until four consecutive sets of readings are within the specified limit.

7.3 Calculations

7.3.1 The rating in Btuh (q_b) shall be obtained by the following:

$$q_b = W\,(h_e - h_L)$$

Where W = weight rate of liquid flow in lbs. per hr.
h_e = enthalpy of liquid entering the unit, Btu/lb.
h_L = enthalpy of liquid leaving the unit, Btu/lb.

7.3.2 The rating in tons (q_t) shall be obtained by the following:

$$q_t = q_b \;/\; 12{,}000$$

7.3.3 For liquids where enthalpy tables are not available as such, the enthalpy difference shall be obtained by the use of the product of the mean specific heat and the temperature difference:

$h_e - h_L = c\,(t_e - t_L)$ where
t_e = average entering liquid temperature, deg F.
t_L = average leaving liquid temperature, deg F.
c = mean specific heat of liquid between temperatures t_e and t_L

APPENDIX

SECTION 8

8.1 *Standard Ratings* shall be given under the condition specified in at least one of the groups listed in Table I.

8.2 *Conditions for Application Ratings*

8.2.1 *Application Ratings* give capacity and performance under operating conditions other than those designated for Standard Ratings. Each set of Application Ratings shall be expressed in the same terms as the Standard Ratings.

8.2.2 *Published Ratings* shall include one or more Standard Ratings identified as such by Group numbers in Table I.

8.3 *Ratings of Liquid Chilling Packages* shall be expressed in the following terms.

8.3.1 The general rating terms shall include all those listed in Section 3.1.1, and in addition the following:

a. Specification of refrigerant used in accordance with ASRE Standard 34-57.

b. Statement of operating refrigerant charge, lbs.

8.3.2 *Ratings for units with water-cooled condensers* shall include all terms listed in sections 3.1.1, 3.2, and 8.3.1.

8.3.3 *Ratings for units with evaporatively-cooled condensers* shall include all terms listed in sections 3.1.1, 3.3, and 8.3.1.

Note: For evaporatively-cooled units, the maximum water supply shown in the ratings may be increased for application purposes when required by water hardness and/or local conditions, and the increased quantity should be specified.

8.3.4 *Ratings for units with air-cooled condensers* shall include all terms listed in sections 3.1.1, 3.4, and 8.3.1.

8.4 Tests for rating purposes shall be conducted in accordance with the previous sections of this standard, and shall include simultaneous determination of additional information sufficient to obtain items listed in section 3.3 and section 8.3 for complete ratings.

TABLE I STANDARD CONDITIONS FOR TESTING

Group No.	Sub Group No.	Water Cooled Condenser: Cooling Water Temp F — In	Out	Evaporatively Cooled Condenser: Air Entering Temp F — Dry Bulb	Wet Bulb	Air Cooled Condenser: Air Entering Temp F — Dry Bulb	Liquid Cooler: Liquid Temp F — In	Out	Liquid Cooler: Minimum Ambient Air Temp F
I	Ia	75	95	95	75	95	56	46	56
	Ib	85	95						
II	IIa	75	95	95	75	95	40	35	40
	IIb	85	95						
III	IIIa	75	95	95	75	95	75	35	75
	IIIb	85	95						
IV	IVa	75	95	95	75	95	20	15	20
	IVb	85	95						

PROJECTED DIAGRAMS LOCALIZE CONTROLS

Troubles in air conditioning or heating systems in a recently remodeled building are pinpointed easily by controls featuring a ground glass projection screen built into the center of a master control panel. The latter regulates the air conditioning, heating, and water temperature for this entire building of Eastman Kodak.

A color photo-diagram of the particular system is projected automatically on the screen. This immediate reference proves an important aid in locating troubles or malfunctions. The master panel controls temperatures of six different kinds of water for drinking, washing, and photographic operations. It shows at a glance the status of cafeteria refrigeration and of cold storage rooms for film. Additionally, the panel provides a double-check by ear. The operator can switch on a microphone in each of the fan rooms and listen in, too.

BULLETINS

Air Conditioning and Refrigeration Products. Catalog No. 82 covers in its 40 pages this manufacturer's complete line of tools and supplies for the refrigeration and air conditioning industries. Full details on refrigerant hose and reusable couplings, Torpedo Driers and "Make-Up" lines which enable the user to make his own charging lines to any length needed are given. Also included are flared and compression fittings and charging and testing units.
Imperial Brass Manufacturing Company, 6300 W. Howard St., Chicago 48, Ill.

Flame Failure. Highlighting safety factors of the unit in eliminating explosions, and pointing up its flexibility in application to oil, coal or gas installations, Bulletin 523 covers working principles in detail, with a diagrammatic sketch of installation, and emphasizes that the flame failure safeguard reacts only to a flickering flame. Of special interest to engineers are the descriptions of the operating principle.
Photomation, Inc., 96 S. Washington Ave., Bergenfield, N. J.

Filters. Consisting of a thin cellulosic membrane containing millions of tiny pores, these filters provide fine filtration for the analysis and purification of liquids and gases. Further extending the advantages of the standard filter to process applications is the new Microweb Filter. 30-page Technical Bulletin 759, descriptive of these filters, contains sections on Basic Characteristics, Principal Applications, Analytical and Process Apparatus, and Field Monitors, among others.
Millipore Filter Corporation, Bedford, Mass.

Plumbing and Heating Products. Data on flow controls, water mixers, vent valves and temperature and pressure relief valves, among other instruments, is included in this catalog.
Dole Valve Company, 6201 Oakton St., Morton Grove, Ill.

Compressed Air Fundamentals. Produced to aid in the selection of a small packaged air compressor for either automotive or industrial application, 16-page Form 1548 describes compressed air, single and two-stage compressors, piston displacement, actual delivery, unloading of compressors, regulation and types of control used. Other material included is information on compressor oils, pipe sizes, wire sizes and terminology and definitions used in connection with the compression of air. Extensive tabular and chart information is given.
Ingersoll-Rand Company, 11 Broadway, New York 4, N. Y.

Low Temperature System. Completely self-contained, ready to plug into an electric outlet, this low temperature system provides temperatures to −100 F or lower with either wet bath or dry storage. As covered in this flyer, the system is said to provide a liquid tight refrigerated area. Capacities from 0.6 to 6.5 cu ft are offered. It may be used for chemical, oil, electronic, industrial, metallurgical and other types of processing and testing.
Remcor Products Company, 321 E. Grand Ave., Chicago 11, Ill.

Make-Up Air Problems. Need for make-up air arises when fans draw fumes, smoke or odors from such buildings as bakeries, chemical process plants, mills and others. Negative pressure created by forced exhaust creates drafts, reverses air flow in flues, and adds considerable load on the heating system. Bulletin EN-5911, 4 pages, describes how make-up air properly introduced and tempered can overcome negative air pressure and eliminate these problems.
Reznor Manufacturing Company, Mercer, Pa.

Refrigerant Data Sheet. Nineteen physical, chemical and performance characteristics are given for five Ucon Refrigerants in 4-page Bulletin F-40408. In addition, a table showing the vapor pressure range from −150 to 200 F is included.
Union Carbide Chemicals Company, Fluorocarbons Div, 30 E. 42nd St., New York 17, N. Y.

Electronic Air Cleaners. Sizing guides, listing rated capacities in cfm at given efficiencies for each model, are now included in this manufacturer's 12-page Catalog 76-4369, for the Selector Series of Electronic Air Cleaners.
Minneapolis - Honeywell Regulator Company, 2747 Fourth Ave. S., Minneapolis 8, Minn.

Packaged Air Conditioners. Dimensions and specifications, as well as detailed construction drawings and other engineering information, are given for the line of packaged air conditioners described in 4-page Catalog No. 571. For commercial and industrial comfort cooling installations, these completely self-contained units may be installed with or without duct work, may be air or water cooled, and come in sizes from 3 to 15 ton.
Acme Industries, Inc., 600 N. Mechanic St., Jackson, Mich.

Sound Insulation. Comprising perforated and fissured wood fiber and mineral acoustical tiles, glass fiber ceiling boards, sound insulating doors and aluminum acoustical systems, this manufacturer's line of sound control products is shown in 12-page Catalog 60. Included are construction details; recommended installation, maintenance and application methods; and official data and ratings.
Elof Hansson, Inc., Acoustical Div, 711 Third Ave., New York 17, N. Y.

Aluminum Windows. Contained in two bulletins, of six (8½ x 11 in.) and eight (3½ x 6¼ in.) pages, respectively, are descriptions of two types of aluminum windows, the Series 1000 and the Z-Bar. Designed with an integral fin serving as either a nail-in flange for wood or veneer or as an anchor flange in masonry, the former features an integral head and sill drip to save installation time. The latter style may be used in applications with tubular mullions and may be set at any angle.
Ventilaire Products Company, Inc., 2431 Harvey St., Muskegon, Mich.

Baseboard Radiation. Features of 5 in 1 Baseboard Radiation include adjustable expansion panels permitting use of standard length units without cutting on the job, a formed inside corner eliminating mitering back panels, arrangements for concealing return piping within the enclosure, and heating element supported on brass rollers for free movement of the tubing in expansion and contraction. Bulletin R-596, 4 pages.
Rittling Corporation, Buffalo, N. Y.

Automatic Controls. Thermostats; controls for cooling towers, duct-fire protection, freeze protection, photographic tanks and water chillers; and various pressure and temperature controls are the subject of 24-page Bulletin 1487-AL. Ratings, specifications and cuts of each item are included.
Penn Controls, Inc., Goshen, Ind.

Air Dehumidifier. Using a liquid absorbent for drying air to control its moisture content, the Hygrol Air Dehumidifier is designed to be used for industrial processing and product dry-
(*Continued on page* 81)

Handling the

Frost factor

in low temperature case design

B. L. HERRMANN
Member ASHRAE

Frozen food and ice cream display cases have undergone many changes in the past few years as the growth of merchandising has demanded more open product display. Horsepower has increased, top openings have become larger and glass areas greater. Heavy frost accumulations on the first refrigerated plates spurred their replacement by forced air or gravity coils. This paper deals with frost and its effects on the design features of the present day forced air type merchandising cases.

The design of almost every component of a low temperature display case is governed to a large extent by frost. Larger compressors are required, coils become blocked, cutting off air circulation, or the product itself takes on an unsalable appearance. Each part is affected individually and all are affected integrally, since an inadequate design in any one part may cause a malfunctioning or completely inoperative case.

Rules to follow in designing around frost might be these four: Keep moist air out of the case as much as possible, control the location and amount of frost buildup, defrost properly and completely, and eliminate frost from objectionable places. If the first could be exploited fully, the remaining three would be unnecessary, but all are still factors to be dealt with.

In designing a new case, the engineer may feel that the compressor is too small for drafty conditions. Actually, a great percentage of lost Btu's is due to the sensible and latent heat required to produce frost from the water vapor of the air. It is possible to reduce the size of the top opening in order to admit less water-laden air, however, this presents a further problem in the resultant reduction of display area. Slanting the front glass is one means of reducing the opening, while still permitting adequate display, and is used widely. Transparent wings, shields, and high back superstructures are sometimes employed to protect the opening from drafts, but are not always practicable because of the styling of the compartments.

Air handling is perhaps one of the more important contributions to a good design, usually yielding only to cut and try methods of design improvement. Air in the product compartment must be kept as quiet as possible, any turbulence drawing in warm moist air from the outside will cause a rapidly blocked coil. Even air distribution must be obtained along the full length of the case to prevent spots of high velocity air at one point and low velocity air at another. In extreme cases there may be reverse air flow. Distribution devices include a high pressure plenum ahead of the discharge grille and baffles ducting to the discharge. One of the more reliable methods is the use of multiple fans spaced over the length of the case and blowing against a spreader baffle.

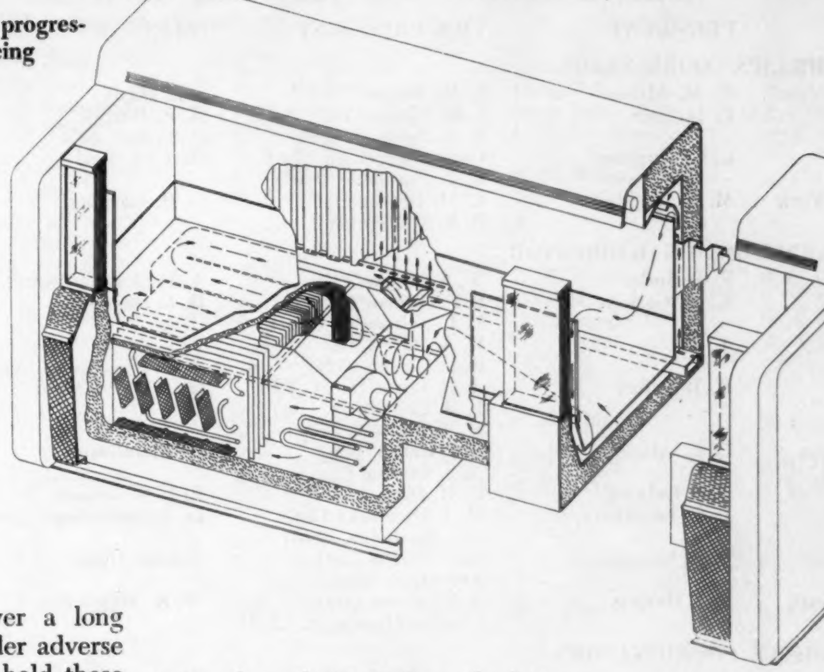
Once having achieved an even distribution of the proper amount of air transgressing the opening, it is imperative that the air stream flow smoothly across in the top layers. A highly workable combination is to diffuse the lower portion of the air stream downward onto the product, with the upper layer holding the air in as a blanket. This can be done with splitters and baffles or by a perforated baffle lowered from the top, diffusing the lower air and allowing the remainder to flow evenly over the top. Air treated in this manner will not only prevent much of the air infiltration, but will keep the upper regions of the case free from frost caused by turbulence.

Air delivery is assisted by discharging and returning the air at as low a level as possible with fairly large grilles. Multiple outlets which allow the air to be discharged and returned at even lower levels as the product load decreases have been found successful for this purpose. With these designs a larger volume of air may be supplied without undue velocities requiring control. The ducting within the case must also produce a smoothly flowing air. This is true particularly of the return ducts, as any amount of disturbance will cause the moisture from this air to be deposited as frost at these points.

The coil construction is a key factor, since it is the logical place for all frost to collect. It must cool the product compartment suffi-

Bernard L. Herrmann is Chief Refrigeration Engineer, Cabinet Div, Anheuser-Busch, Inc. This paper was presented at the Commercial Refrigerating Conference at the ASRE 45th Semiannual Meeting, New Orleans, La., December 1-3, 1958.

Fig. 1 Coil with progressively closer fin spacing



ciently and hold it over a long period of time even under adverse conditions. In order to hold these temperatures well into the subzero range for extended periods, a means must be found to prevent coil blockage. One method is based upon a coil of extremely large face area, enabling it to accumulate a great deal of frost before blocking occurs. Another is the use of two coils separated and ducted in such a way that one or both are used in a recirculating manner. In this design a certain percentage of air is passed through the coil a second time to dehydrate and cool it further. The disadvantage of these methods is the large area required for housing the coils.

A third design, one of the earlier types which is shown in Fig. 1, uses one or more smaller coils of progressively closer fin spacing. The wider fin spacings are used mainly for dehydration of the moist return air, while the close spacings near the coil outlet remove the sensible heat of the dry air. An air gap is employed between the various spacings to minimize blockage at the entrance to the narrower spacings. The principle of this coil also applies to two coils of different fin spacings with a recirculation of air through the one of wider spacing.

The location of coils is important in that the air leaving the coil should be as close to the discharge grille as possible. As much as a 10 deg loss is normal for air going through ducts from the bottom of the case to the discharge at the top. Since the coldest air possible is required at the product, this 10 deg loss can be serious. Care must be observed, when locating a coil in this manner, to prevent the return air ducts from removing too much of the moisture before it reaches the coils, thus blocking air circulation. If two coils are used, one can be located remotely from the other, thereby collecting a major portion of the frost and keeping the ducts clear.

When these points have been accomplished, the task of defrosting is simplified. A defrost must be complete and rapid. This includes ducting and complete drainage of water, as well as the coil. The method of defrost, whether electrical or hot gas, will not be discussed here. Both have advantages and disadvantages, but one type may be more suited to a particular case design than another. Since ducts must also be defrosted, good heat coverage is essential, and concentration of heat is to be avoided. The circulating fan is often kept in operation for this purpose.

Determination of the frequency of defrosting should be done by field testing, since it is almost impossible to duplicate field conditions in a test room. Defrosting should be initiated at a time when air circulation is first effected. At the present time defrost periods are initiated at time intervals. Because climatic conditions vary the frequency required for defrosting, it has not been possible to obtain the optimum, and frequency must therefore be geared to the more severe conditions. More important to remember, however, is that a coil which is choked off will no longer keep the product temperature. A coil blocked off for a few hours before a defrost is scheduled may cause serious consequences, since heat is added to an already warm product. Generally, it is better to defrost too frequently than too infrequently.

Competition and economics often dictate compressor sizing. It is therefore imperative that every consideration be given to case design in order to obtain maximum efficiency. Frost is a big factor in wasted Btu's, for every pound of water vapor converted to frost, approximately 1100 Btu's are consumed. This figure added to the inefficiencies of defrosting represents a staggering percentage of the compressor's capacity.

Frost buildup within the product compartment may occur because of particular design or from draft conditions after installation. The usual manner of prevention is

(*Continued on page* 114)

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*term as Regional Representative extended by Board to January 1960

Meetings ahead

October 5-7 – American Gas Association, 41st Annual Convention, Chicago, Ill.

October 30-November 2 – Refrigeration Service Engineers Society, Annual Convention, Atlantic City, N. J.

November 1-2 – Air-Conditioning and Refrigeration Wholesalers, Annual Meeting, Atlantic City, N. J.

November 2-5–11th Exposition of the Air-Conditioning and Refrigeration Industry, Atlantic City, N. J.

November 9-13 – National Electrical Manufacturers Association, Annual Meeting, Atlantic City, N. J.

November 9-13 – Institute of Boiler and Radiator Manufacturers, Semiannual Meeting, Absecon, N. J.

November 17-19 – Building Research Institute Semiannual Conference, Washington, D. C.

December 3-4.– National Warm Air Heating and Air Conditioning Association, Annual Convention, St. Louis, Mo.

December 26-31 – American Association for the Advancement of Science, Annual Meeting, Chicago, Ill.

February 1-4 – American Society of Heating, Refrigerating and Air-Conditioning Engineers, Semiannual Meeting, Dallas, Texas.

February 1-4–2nd Southwest Heating and Air-Conditioning Exposition, Dallas, Texas.

April 27-30 – 3rd Western Air-Conditioning, Heating, Ventilating and Refrigeration Exhibit and Conference, Los Angeles, Calif.

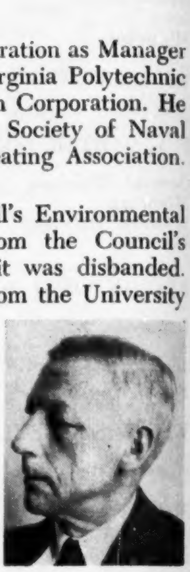
May 1-4 – Air-Conditioning and Refrigeration Institute, Annual Meeting, Hot Springs, Va.

May 19-21 – Refrigeration Research Foundation, Annual Meeting, Denver, Col.

June 13-15–American Society of Heating, Refrigerating and Air-Conditioning Engineers, 67th Annual Meeting, Vancouver, B. C.

People

Charles M. Hull has joined Bohn Aluminum and Brass Corporation as Manager of the Industrial Coil Dept. An engineering graduate of Virginia Polytechnic Institute, Mr. Hull was formerly Sales Manager with Aerofin Corporation. He is a member of the American Society of Naval Engineers, Society of Naval Architects and Marine Engineers and National District Heating Association.



Thomas Bedford, director of the Medical Research Council's Environmental Hygiene Research Unit in London, England, retired from the Council's staff on September 20th, at which date the Research Unit was disbanded. Dr. Bedford, who holds the degrees of D.Sc. and Ph.D. from the University of London, has been engaged in studies connected with occupational health for nearly 40 years, and is a special lecturer in Industrial Hygiene, London School of Hygiene and Tropical Medicine. A prolific writer, he is author or co-author of numerous technical papers, reports and bulletins published in England and in continental periodicals, and of several books, including "Basic Principles of Ventilation and Heating". He is perhaps best known in this country for his work on the development of the Equivalent Warmth Index, which takes into account the radiant temperature of the environment as a factor in comfort reactions. In 1957, Dr. Bedford was awarded the French Gold Medal in recognition of his work on the science of artificial climate.

Samual B. Paschal is now in charge of the J. P. Ashcraft Company, Inc., Oklahoma City office, which has taken over Baltimore Aircoil Company products in the state of Oklahoma.

Ralph D. Moore, appointed Refrigeration Sales Representative of Ansul Chemical Company, will cover the state of California from Fresno to the northern boundary and Reno, Nev.

Ronald S. Rose, an engineering graduate of Duke University, has been appointed Sales Manager of Canadian Ice Machine Company, Ltd. He was formerly a sales manager with York Corporation.

Henry R. Krueger, a project engineer for the past two years, has been named to the post of Director of Engineering of Acme Industries, Inc. A 1949 graduate of the University of Oklahoma with a Bachelor of Science degree in mechanical engineering, Mr. Krueger joined the firm in 1957, after several years of service with Governair Corporation, where he was Chief Engineer.

Frank D. Klein, formerly Chief Engineer for Governair Corporation, is now Product Manager of air conditioning products for Dunham-Bush, Inc. A member of the American Society for Testing Materials and the American Association for the Advancement of Science, Mr. Klein is the author of a book, "Air Conditioning and Distribution Requirements in Year Round Systems", and of several technical articles.

James E. Williams will be representing Air Coils Manufacturing Company, Ltd., and Dole Refrigerating Products, Ltd., in the provinces of British Columbia and Alberta. Through his association with J. & E. Hall and Canadian Ice Machine, Mr. Williams has been connected with the refrigeration industry for 20 years.

Charles R. Kohlenberger has been elected to the position of President of Kohlenberger Engineering Corporation, following the death of its founder and President, **Hans H. Kohlenberger**. An engineering graduate of the University of California, he had held the post of Vice President of Engineering.

Joseph A. Galazzi has been named Vice President in charge of the Foreign Operations Div of Tecumseh Products Company, having served since 1955 as Chief Engineer of Commercial Refrigeration. Previously he was Chief Engineer for International Harvester Refrigeration Div, and had been with York Corporation for 12 years prior to that, starting as a student engineer and rising to Assistant Director of the Engineering Laboratory. A mechanical engineering graduate of Tufts University, he has been Secretary and Vice Chairman of the Evansville Section of the former ASRE, is the holder of several patents in the refrigeration field and has written several technical papers for Refrigerating Engineering. Prior to his new appointment, he was most active in the field of design and development.

R. C. Binder, formerly Professor of Mechanical Engineering at Purdue University, now holds that same position in the Div of Industrial Research, Washington State Institute of Technology, State College of Washington. Prof. Binder is the author of a paper, "Some Methods for Investigating Noise From Compressors Used on Household Refrigerators", presented at the Lake Placid meeting of ASHRAE.

John L. Adams has been appointed Sales Engineer of Sporlan Valve Company, and will work with the Atlanta Field Sales Office. An engineering graduate of California State Polytechnic College, he has worked in the company's Sales and Engineering Departments prior to this appointment.

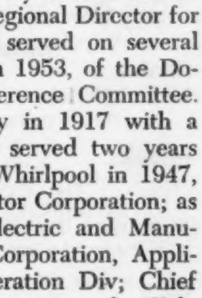
J. Fraser Morris, General Sales Manager for Universal Cooler Company, Ltd., retired on June 1st after 31 years with the firm. Mr. Morris is a past President of Commercial Refrigerator Manufacturers Association.

R. W. Ayres retired on August 1st from the St. Paul Div of Whirlpool Corporation, where he has acted as Engineering Consultant for the past two years, after having served for ten years as Chief Engineer. Identified with the refrigeration industry since 1919, Mr. Ayres is a former member of ASRE Council, and was a Regional Director for the term ending June, 1956. He has served on several ASRE committees and as Chairman, in 1953, of the Domestic Refrigerator Engineering Conference Committee. After graduation from Yale University in 1917 with a degree in mechanical engineering, he served two years with the U. S. Army. Before joining Whirlpool in 1947, he served as Chief Engineer at Coolerator Corporation; as Director of Research for Sunbeam Electric and Manufacturing Company; Chief Engineer at Stewart-Warner Corporation, Appliance Div; as a staff engineer at General Electric Refrigeration Div; Chief Engineer for Savage Arms Refrigerating Div; and as an engineer for Yale and Towne Manufacturing Company.

Thomas L. Valentine, formerly Air Conditioning and Heating Engineer with Samuel L. Burns and Associates, Inc., is a partner in a newly formed firm, Ragon and Valentine, Consulting Engineers, formed for the practice of electrical and mechanical engineering.

Harold W. Freeborn, an executive officer of Hussmann Refrigerator Company, Ltd., since it acquired the assets of Ruddy-Freeborn Company, a firm of which he was a vice president, has been named to the newly created post of Senior Vice President.

H. C. Diehl has been elected to Honorary Life Membership in the Refrigeration Research Foundation, according to an announcement by the Board of Governors. Award of the Certificate of Honorary Membership will be made at an Executive Committee Meeting Dinner on October 24 at the La Salle Hotel, Chicago, Ill. Mr. Diehl, an active contributor to committees and publications of the former ASRE, retired on December 31, 1958 as the Foundation's only director since its formation January 1, 1944.



BULLETINS

(*Continued from page 75*)

ing at low temperatures and the air conditioning of spaces for packaging and storage of moisture-sensitive materials. Operating description and control method, together with a chart of dehumidifying capacities at various temperatures and cfm of air handled, are included in 4-page Bulletin No. 140.
Niagara Blower Company, 405 Lexington Ave., New York 17, N. Y.

Soldering Guide. Intended to fill the need for information about solder preforms and their use, 8-page booklet "A Guide to Preform Soldering" gives data on the range of preform shapes, the alloys from which they are made and their use in automated production processes. Also described are heating methods, flux selection, metals characteristics and alloy selection.
Alloys Unlimited, Inc., 21-01 43rd Ave., Long Island City 1, N. Y.

Cold Weather Testing. Located on Mt. Washington, N. H., are large facilities for cold weather testing of materials and equipment used in the construction industry. Weather conditions are cited as approximating those of the Arctic. These facilities, detailed in a 20-page bulletin, are such that clients may, if desired, carry on work with their own personnel.
Gorham Laboratories, Inc., Gorham, Me.

Automatic Controls. Listing this manufacturer's entire line of automatic controls for heating, refrigeration and air conditioning, 56-page Catalog Unit R-1650 contains many new products. Full specifications, plus a description of operation and general applications are included in each product listing. The catalog is cross-indexed for easy location of controls by type or use.
White-Rodgers Company, 1209 Cass Ave., St. Louis 6, Mo.

Tube Fittings. Headings of sections in this 20-page catalog include How to Order, "O-Ring" Seal Hydraulic Tube Fittings, Straight Thread Fittings, Two or Three Piece 37 Deg Flare Tube Fittings, and Hydraulic Accessories. Each fitting in the line is illustrated, and a list is given of all distributors stocking these products.
Lenz Company, Dept. 131-A, 3301 Klepinger Rd., Dayton 16, Ohio.

Candidates for ASHRAE Membership

Following is a list of 74 candidates for membership or advancement in membership grade. Members are requested to assume their full share of responsibility in the acceptance of these candidates for membership by advising the Executive Secretary on or before October 31, 1959 of any whose eligibility for membership is questioned. Unless such objection is made these candidates will be voted by the Board of Directors.

Note: * Advancement † Reinstatement

REGION I

Connecticut

CANNISTRARO, J. C., Sales Engr., Connecticut Valley Refrigerating, Inc., E. Hartford.
HERRINGTON, L. P., Dir. of Research JBPF & Lecturer in Envir. Physiol. YMS., John B. Pierce Foundation & Yale University, New Haven.
KULAS, E. F., JR., Com. & Indus. Sales Engr., Hartford Gas Co., Hartford.

Massachusetts

ROCHFORD, W. A., Htg. & A-C. Cons., Western Mass. Electric Co., Springfield.

New Jersey

FEHRENBACH, R. J., Dir. of Engrg., Thermo Equipment Corp., Newark.
MONROE, L. H., Treas., All State Air Conditioning Co. of N. J., Inc., Newark.

New York

BELSKY, G. A.,† Exec. Vice-Pres., Air Conditioning, Inc., Ossining.
CARETSKY, MARVIN,* Partner, W. A. DiGiacomo Associates, New York.
CARR, J. E., Maint. Foreman III, Port of New York Authority, Jamaica, N. Y.
LOWELL, J. M., Power Plant Foreman, Port of New York Authority, Jamaica, N. Y.
SEIFERT, D. J.,* Sales Engr., American Standard, Industrial Div., Buffalo.

Canada

BARRANCE, D. A., Supt. Cust. Service, Provincial Gas Co., Ltd., Fort Erie, Ont.
BRACE, R. C., Sales Engr., Honeywell Controls, Ltd., Hamilton, Ont.
CALDWELL, F. H., Dist. Mgr., Honeywell Controls, Ltd., Hamilton, Ont.
ELLIOTT, G. E., Mech. Engr., H. K. Walter & Assoc., Hamilton, Ont.
REDDICK, J. W., Designer, McGregor & Beynon, Ltd., Toronto, Ont.
SCHOCK, R. E.,* Sales Engr., Canadian Blower & Forge, Toronto, Ont.
WOROBEY, V. E., Sales Engr.. Johnson Controls, Ltd., Regina, Sask.

REGION III

Delaware

PALCZEWSKI, T. T., Supvsr., Robert P. Schoenijahn, Cons. Engr., Wilmington.

Maryland

BROWN, P. J., Gen. Mgr., Industrial Sheet Metal, Inc., Baltimore.
SIMONINI, K. D., Maint. Engr., Great Atlantic & Pacific Tea Co., Baltimore.

Pennsylvania

LACY, F. P.,* Assoc., Lacy, Atherton & Davis, Wilkes-Barre.
SHEEDER, J. T., Sales Engr., National U. S. Radiator Corp., Pittsburgh.

Virginia

MUNIER, R. A., Designer, Newport News Shipbuilding & Dry Dock Co., Newport News.
SANDERS, J. L., Chief, Environmental Engineering Br., QM Research & Engrg. Field Evaluation Agency, Fort Lee.

REGION IV

North Carolina

MCKNIGHT, H. F.,† Mech. Design Engr., Douglas Aircraft Co., Inc., Charlotte.
JOHNSON, J. H., Repr., Armstrong Cork Co., Charlotte.

REGION V

Illinois

HEUMANN, H. C., Sales Mgr., Illinois Electric Works, Inc., East St. Louis.

Indiana

AHLF, R. E.,* Prod. Engr., Whirlpool Corp., Evansville.
PHILLIPS, D. S.,* Prod. Supvsr., Whirlpool Corp., Evansville.

Ohio

KELLY, A. L., Sales Engr., General Electric Co., Columbus.
O'BRIEN, C. E., Owner, C. E. O'Brien Co., Dayton.
SAMPSON, J. N.,† Mgr., Lumm Corp., Toledo.
STAHR, P. H., Part Owner, C. E. O'Brien Co., Dayton.

REGION VI

Illinois

HOSLER, A. D., Chief Mech., F. R. Valvoda, Cons. Engr., Oak Park.

Michigan

SCHILKEN, D. R.,† Vice-Pres. & Sales Mgr., Acme Insulations, Inc., Grand Rapids.
WISNER, R. R., Designer, American Motors Corp., Detroit.

Wisconsin

BARTH, R. M., Field Engr., Johnson Service Co., Milwaukee.

REGION VII

Alabama

WERSBA, M. D., Designer, Engr., Project Engineers's, Birmingham.

Louisiana

WILSON, T. W., Sales Engr., Airtemp Div., Chrysler Corp., New Orleans.

Missouri

VERMA, S. M., Engr., Marlo Coil Co., St. Louis.

Tennessee

ALESSIO, V. H., Supt., Evans-Hailey Co., Inc., Nashville.
CARNEY, W. H., Estimator, M. T. Gossett Co., Nashville.
CASTEEL, W. J., Technician, Johnson Service Co., Nashville.
DAHLINGER, F. W., JR., Pres., Climate Control Co., Nashville.
DWYER, S. E., JR., Br. Mgr., Armstrong Contracting & Supply Corp., Nashville.
EARLY, H. F., Owner, Air Filter Sales & Service Co., Nashville.
FARRIS, G. F., Sales Engr., Carrier Corporation, Nashville.
HARWELL, R. H., Partner, Harwell & Rosselot, Nashville.
KELTNER, M. C., JR., Field Repr., Crane Co., Nashville.
MASSA, C. T., JR., Partner, John E. McCluen Co., Cookeville.
NEBLETT, E. R., Designer, F. Kurzynske, Cons. Engr., Nashville.
NORRIS, R. G., Estimator, John Bouchard & Sons Co., Nashville.
O'BRIEN, R. C., †* Engr., Howard Nielson & Lyne, Inc., Nashville.
PARKES, J. L., Assoc., I. C. Thomasson & Assocs., Nashville.
SPALDING, J. H., Repr., Young Sales Corp., Nashville.

REGION VIII

Arkansas

MILLER, J. D.. Sub Office Mgr., Trane Co., Little Rock.
OXLEY, R. S.. Technician, Johnson Service Co.. Little Rock.
WOODSMALL, W. M., Engr., Arkansas Louisiana Gas Co., Little Rock.

Texas

PURDY, J. M.,* Partner, Jessen, Jessen, Millhouse & Greeven, Austin.

REGION IX

Utah

SONNTAG, G. T.,† Sales Engr., American Standard, Industrial Div., Salt Lake City.

REGION X

Arizona

LA GASSE, C. D.,* Pres., Thomas Heating & Air Conditioning, Inc., Phoenix.

Others are saying—

that . . . . . . designed on the basis of a new idea, a unit originally introduced for the dehydration of condensers and evaporators is being used for dehydration of compressed air, its effect cited as being greater than ordinary vacuum drying. A constant supply of compressed air is obtained with a saturation point of —40 F (—40 C). Acting effectively under heavily fluctuating surrounding temperatures (—22 F (—30 C) to 104 F (40 C)), the unit uses Refrigerant CHF_2Cl at an evaporating pressure of from 0 to 0.14 atmosphere (approximately —40 F (—40 C)). *Danfoss Journal, British Edition, 2nd Quarter 1959 No. 2, p 24.*

that . . . . . . at sessions of the 10th International Congress of Refrigeration, held August 18-26 in Copenhagen, Denmark, presentations were made of more than 200 technical papers. Abstracts of the papers of Commissions II, III and IV are contained in the Special Congress Supplement of this issue. *Modern Refrigeration and Air Control, August 1959, p 1 (British).*

that . . . . . . newly built air conditioned homes are being roofed in white and pastel colored asphalt shingle or white granule plastics coatings, as light colored roofs have been found to aid air conditioning, reflecting a large percentage of the sun's rays and thereby reducing the load on the air conditioning unit. *National Roofer, July 1959, p 13.*

that . . . . . . study of the possibility of installing a high temperature water system should be made in cases where old boiler plants or pipe lines are being replaced or in large new projects requiring considerable heat distribution, as this system has many advantages over steam. It should be borne in mind that existing steam systems in specific areas may be retained as part of a high temperature water system by use of heat exchangers. *Hawaii Industry, June 1959, p 29.*

that . . . . . . application of solar energy in refrigeration and air conditioning is primarily through use of absorption refrigeration machines in which the heat required for operation is provided either by concentrating the sun's rays directly onto the generator of the apparatus or by supplying a solar-heated fluid to transfer heat in the generator. Methods of collecting solar energy involve use of either a flat plate absorber, cylindrical parabolic concentrator, paraboloidal concentrator or plane mirror concentrator. Several systems are cited as operating with varying degrees of efficiency, the method being more suitable to tropical climates where the amount of sunlight is fairly constant than in regions where the majority of days are cloudy. *Journal of the Institution of Heating and Ventilating Engineers, July 1959, p 97 (British).*

that . . . . . . optical density methods of estimating domestic smoke emission gravimetrically and comparison of these methods with direct gravimetric measurement of the smoke emission from a number of different types of domestic appliance indicate there is no fixed relation between smoke concentration and optical density. The ratio of the true weight of smoke to that estimated by the optical method varies widely under different conditions. Light scattering effects at high concentrations are cited as possible reasons for the unreliability of optical measurements. *Journal of the Institution of Heating and Ventilating Engineers, July 1959, p 108 (British).*

that . . . . . measurements taken with Eppley pyrheliometers suspended on the north and south vertical surfaces of a tower, approximately 26 ft above the ground surface, indicated that solar radiation received by the surfaces can be separated into two classes depending upon the season of the year. Between the vernal and autumnal equinoxes, radiation on the north wall consists of direct and of diffuse sky and ground reflected components; between the autumnal and vernal equinoxes it consists entirely of diffuse sky and ground reflected components. The converse of this is true of the south wall. For the north surface, value of radiation for the majority of days lay between 0 and 100 Langleys, while the largest number of days on the south surface were in the 200 to 300 Langley class. On only two days of the year did radiation on the north wall exceed 200 Langleys, as opposed to 228 days for the south wall. *Air Conditioning, Heating and Ventilating, August 1959, p 64.*

that . . . . . suggestions offered for achieving the utmost economy in the design and use of a heat pump are: provide industrial air source heat pump systems with not more than four fins per in. to limit frequency of defrost, shown to be proportional to fin spacing; provide central ventilation supply system to the building or to each zone rather than at the individual room unit; select air handling unit coils for the summer requirement to provide design dry bulb and relative humidity requirements at the heaviest summer load condition; program supply water temperature inversely with the outside air temperature to provide water

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California

ALEXANDER, BERNARD, Pres., C. S. Bigelow & Assoc. Inc., Los Angeles.
BROWN, W. L., JR., Pre-Sales Engr., York Corp., Div. of Borg Warner Corp., Los Angeles.
MORROW, A. M., Design Engr., Levine & McCann, Los Angeles.
RUSHER, GEORGE,* Pres. & Chief Engr., Rusherheat Inc., Inglewood.
SONANDER, P. O., Engr., Skidmore Owings & Merrill, San Francisco.
WARREN, J. A., Sales Engr., Coast Heating & Air Conditioning, Los Angeles.

FOREIGN

Australia

HODGE, H. B.,* Dir. & Mgr., Gardner & Naylor Pty., Ltd., Hawthorne, Melbourne, Victoria.

England

SCHOLEFIELD, D. A., Joint Managing Dir., Hargreaves (West Riding) Ltd., Yorkshire.
STRAUSS, J. G., Service Mgr., Temperature Ltd., Fulham, London.
YUAN, J. S., Research & Dvlpt. Engr., Refrigeration Dept., Pressed Steel Co., Ltd., Oxford.

Greece

KARANASSOS, A. C., Gen. Mgr., Cooling & Electrical Laboratory Apparatus, Athens.

South Africa

KING, L. S., Estimating & Design Engr., Airco Engineering Ltd., Durban.

sufficiently warm but not overheated; provide compound compression at lower outside air temperatures when the pumping differential across the compressors becomes high; and subcool the liquid refrigerant to increase the refrigeration effect of each pound circulated to the evaporator thereby to reduce the quantity of refrigerant that the compressor must handle to absorb a given amount of heat. *Air Conditioning, Heating and Ventilating, August 1959, p 69.*

that . . . . . . fast freezing is an important factor in the smoothness of commercial ice cream and derivative products. After initial processing, the mixture, in this plant, is cooled at 40 F in overhead coolers. From there it goes into holding tanks where the temperature is kept constant at 36 F until the ice cream is ready to be packaged. The final step is the piping of the mixture into the fast freezers, each of which can freeze 150 gph. *Southern Dairy Products Journal, July 1959, p 40.*

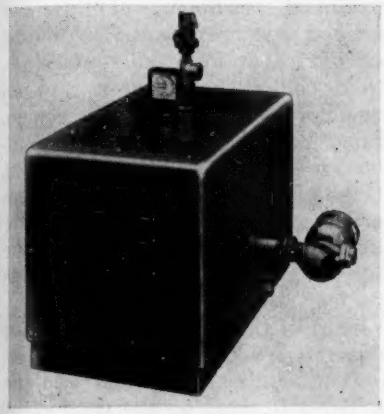
that . . . . . . at the Australian Institute of Refrigeration Conference in Brisbane, Australia, a series of eleven lectures was presented. Dealing with such subjects as cold storage, food preservation and air conditioning equipment, the lectures are reproduced, with some condensation, in this issue. *Refrigeration Journal, July 1959, p 12 (Australian).*

that . . . . . European air conditioning machinery and distribution systems differ in several respects from typical American installations: sensibility to air movement in occupied space limits velocity to no more than ½ fps, and standards for admitted noise level are lower than in America; maximum admitted noise level for an air conditioning system in Europe usually is set at three decibels, with a background noise level of 35 decibels. Preference is given to air and water distribution systems operating at relatively low velocities, usually 6 to 7 fps for water, 1200 fpm for low velocity air and 3600 fpm for high velocity air distribution. The semicentral system is most used for skyscraper air conditioning, with main units placed in the basement and distribution units and registers in the individual rooms. *Consulting Engineer, August 1959, p 92.*

PARTS AND PRODUCTS

HOT WATER BOILER

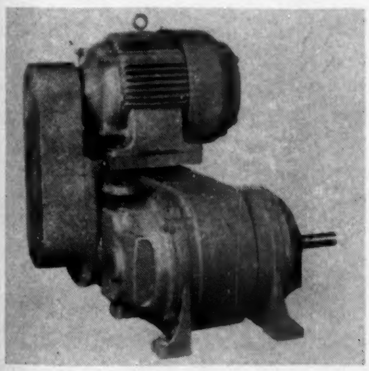
Supplying hot water at temperatures ranging from 60 to 200 F, the Electric Hot Water Heating Boiler is designed for radiant panel, convector, baseboard and radiator type heating



systems and can be used with water chiller for year-round air conditioning. Available in sizes from 40,948 to 2,047,200 Btu, it is suitable for homes, commercial buildings and swimming pools.
Precision Parts Corporation, 400 N. 1st St., Nashville 7, Tenn.

VARIABLE SPEED DRIVE

Featuring separate motor construction and horizontal assembly, this mechanical variable speed drive is cited as offering a wide selection of output speed ranges through proper



selection of v-belt and timing belt. Specifications include ½ to 30 hp, output speeds from 4660 to 1.2 rpm and speed variations from 2:1 to 10:1.
Sterling Electric Motors, 5401 Telegraph Rd., Los Angeles 22, Calif.

AIRBORNE REFRIGERATION

Light weight and compact, these ½ and ¼-hp units were developed for cooling applications in aircraft galleys and related equipment. Available both horizontal and vertical-mounted, the units contain a hermetically sealed compressor, a condenser with co-axial receiver tank and a fan mounted between condenser and compressor, and are completely self-contained types designed for operation on 3-phase, 400 cycle, 208 volt power, standard for turbo prop and jet airliners.
Task Corporation, 1009 E. Vermont Ave., Anaheim, Calif.

HUMIDISTAT

Serving the ventilation of an electrically heated home, the KAH-180 humidistat was added to this line of kitchen ventilating equipment. Featuring a 150 strand human-hair control element, the device is intended to provide accurate settings for controlling excess humidity from 10 to 100% relative humidity. A single-pole, double-throw, line or low voltage ac or dc pilot control, the unit is provided with a tension release device which prevents the hair strands from being damaged by extreme humidity or mechanical strain.
Stewart Industries, Inc., 320 E. St. Joseph St., Indianapolis 2, Ind.

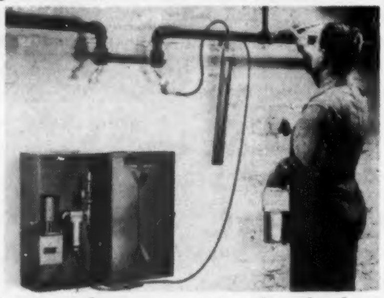
WARMING BOX

Obtainable in two wattages, this continuous duty, gravity connection, in-

dustrial warming box features a "Folded-and-Formed" heating element at 1500 watt and strip heaters at 800 watt. One-in. insulation and thermostatic control are also provided for the box, which operates on 120 volt, single-phase.
Trent, Inc., 201 Leverington Ave., Philadelphia 27, Pa.

GAS LINE TESTING DEVICE

Self-contained and weighing only 24 lb, the unit is a small, electrically operated compressor which builds up and maintains a constant test pressure in low and intermediate pressure gas lines, to make leak detection



faster and more positive than by hand methods. Since pressure is maintained indefinitely without attendance by an operator, the unit makes possible checking of the entire line by one man, as shown.

Adjustable range of the compressor is from 6 ft water column to 5 psi. Switching the selector permits the unit to deliver a maximum of 10 psi.
DeVilbiss Company, Toledo 1, Ohio.

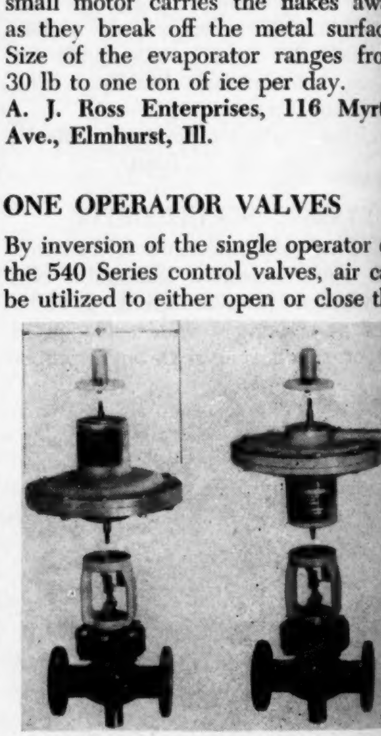
ICE-FLAKE MAKER

Horizontal positioning of this unit is cited as opening new product possibilities in ice-making machines, liquid chillers, refrigerators and refrigerator cars. With this design, drains for melted ice can be eliminated by putting the ice-making parts at the bottom of the storage compartment, and any melted ice-water will return to the coldest part of the assembly, to automatically be made into ice-flakes.

Heart of the device is a cylindrical evaporator, with the refrigerant expanding inside the cylinder and the ice flakes forming on the outside. A helical harvesting spring driven by a small motor carries the flakes away as they break off the metal surface. Size of the evaporator ranges from 30 lb to one ton of ice per day.
A. J. Ross Enterprises, 116 Myrtle Ave., Elmhurst, Ill.

ONE OPERATOR VALVES

By inversion of the single operator on the 540 Series control valves, air can be utilized to either open or close the

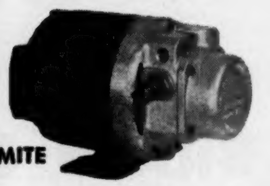


valve. Standard body pressure rating is 300 psi, body temperature rating is 450 F with standard bonnet, and operator temperature rating is 180 F max. Flow coefficient ranges from 0.002 to 13.2.
George W. Dahl Company, Inc., 86 Tupelo St., Bristol, R. I.

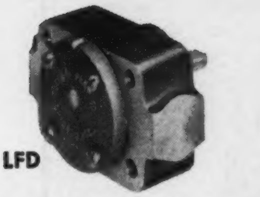
FIVE-TON HEAT PUMP

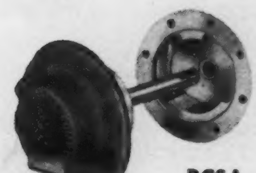
Added to the Weathertron line of air conditioning equipment is a 5-ton, split-system heat pump, consisting of

(*Continued on page 99*)

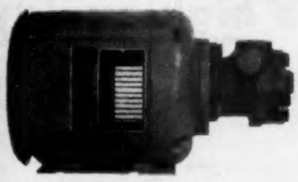

POWERMITE

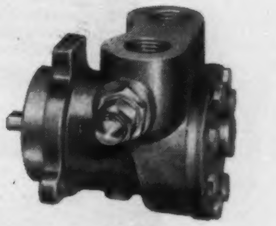

RCK


LFD


RCSA

For your pump problems . . . over 800 different TUTHILL pumps


SERIES 42


LPFV

- **Capacities to 200 GPM: pressures to 1500 PSI**
- **For lubrication, coolant, oil burning, circulating, and hydraulic applications**

For over 30 years the Tuthill Pump Company has been meeting the pump needs of American industry. In literally thousands of demanding applications . . . in lubrication, hydraulics, oil transfer and a wide variety of other services . . . Tuthill pumps are providing the dependable, trouble-free performance which has made them an industry standard.

With over 800 different models Tuthill provides a wide selection. Skilled application engineers, especially trained to "fit the pump to the problem", provide valuable design assistance in precisely meeting your pump requirements.

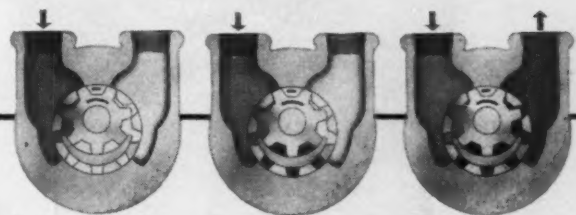
Most Tuthill units employ the time-tested internal gear operating principles described at the right. The complete Tuthill line also includes internal spur gear and sliding vane models.

Many options and modifications

Tuthill pumps can be furnished to fit the requirements of your particular application. For example they can be supplied:

- With or without built-in relief valve
- With automatic reversing feature where pump must be driven from a reversing shaft . . . or a machine must be shipped without knowing ultimate direction of driving unit
- As stripped models to be built into your equipment
- With a wide variety of porting arrangements
- With special shaft seals for various applications
- With provisions for steam jacketing
- With many shaft modifications for drive connections

In short, if your specifications lie within 200 GPM capacity, pressures to 1500 PSI, and speeds to 3600 RPM, Tuthill probably has the answer.


1 2 3

Internal gear pumping principle

In Tuthill internal gear pumps there are only two moving parts. The principle is based on the use of a rotor, idler gear and a crescent shape partition cast integral with the cover.

Power applied to the rotor is transmitted to the idler gear with which it meshes. The space between the outside diameter of the idler and the outside diameter of the rotor is sealed by the crescent. As the pump starts the teeth come out of mesh increasing the volume. This creates a partial vacuum, drawing the liquid into the pump through the suction port (Fig. 1). The liquid fills the spaces between the teeth of the idler and the rotor and is carried past the crescent partition through the pressure side of the pump (Fig. 2). When the teeth mesh on the pressure side the liquid is forced from the spaces and out through the discharge port (Fig. 3).

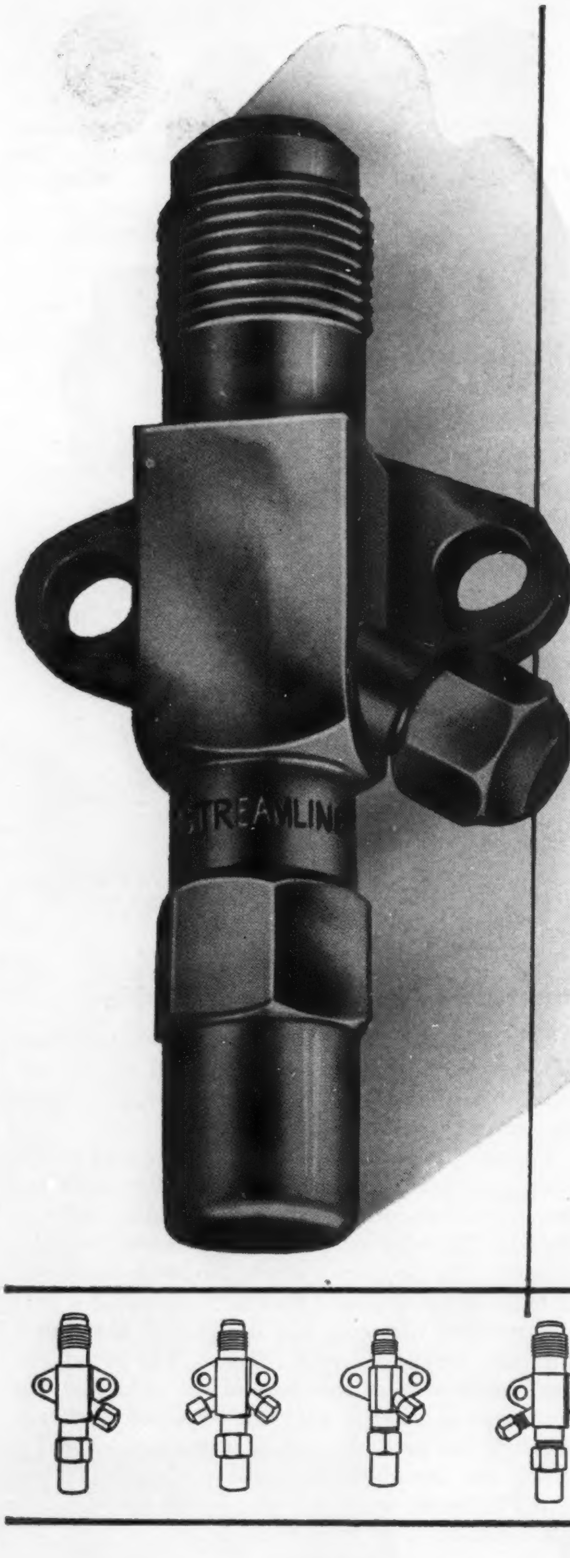
Write today for catalogue 100. Or better yet, ask that a Tuthill Application Engineer call to discuss your specific pumping problem.

Tuthill manufactures a complete line of positive displacement rotary pumps in capacities from 1 to 200 GPM; for pressures to 1500 PSI; speeds to 3600 RPM.

PUMPS FOR YOUR PURPOSE

TUTHILL PUMP COMPANY

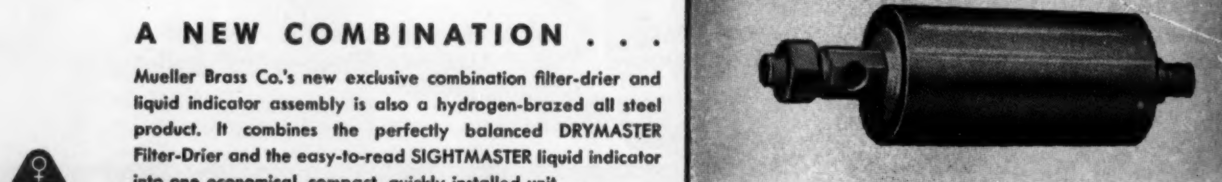
969 East 95th Street, Chicago 19, Illinois



MUELLER BRASS CO.

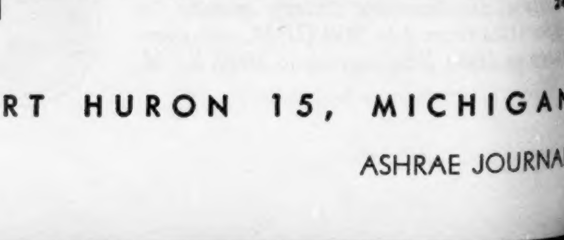
introduces . . . hydrogen brazed steel compressor valves that combine high quality with low price

A new line of Streamline® Hydrogen brazed steel compressor valves is now available to you at amazingly economical prices. They incorporate the same high quality construction as the forged brass valves developed and introduced by The Mueller Brass Co. many years ago. The combination of steel construction, and new Hydrogen Brazing techniques has made possible the production of sturdy, dependable valves that will give outstanding performance in your products, yet hold down those all-important costs.
Included in this new line of valves are compressor valves, (single or double port and adjustable), steel evaporator liquid valves and receiver valves. All are available in both flare and solder type connections. Write us for full information on cost and engineering specifications . . . a Mueller Brass Co. Sales Representative will be glad to call on you.



A NEW COMBINATION . . .

Mueller Brass Co.'s new exclusive combination filter-drier and liquid indicator assembly is also a hydrogen-brazed all steel product. It combines the perfectly balanced DRYMASTER Filter-Drier and the easy-to-read SIGHTMASTER liquid indicator into one economical, compact, quickly installed unit.



MADE IN U.S.A. TO THE STANDARDS OF AMERICAN INDUSTRY

MUELLER BRASS CO. PORT HURON 15, MICHIGAN

(*Continued from page* 94)

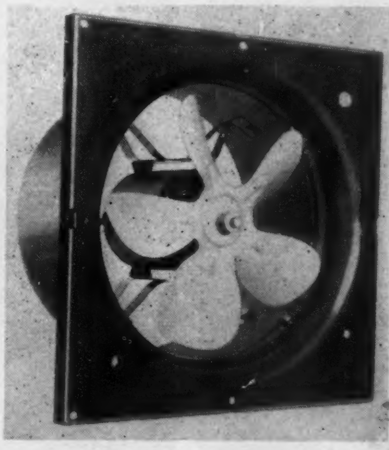
an outdoor remote unit, Model WTA-60B, and an indoor air handling unit, Model WTE60AC. Both heating and cooling without fuel or water, the air handler is designed for all-year operation, and utilizes a 115/230 volt blower motor.

Although furnished in upflow form, the indoor unit can easily be converted to downflow operation and can be used in cooling only applications as well as for heat pump use.

General Electric Corporation, Central Air Conditioners Dept., Tyler, Tex.

TUBEAXIAL FAN

Recommended for use in stationary commercial equipment where shock and vibration are negligible, air moving unit No. 89B222 may be used for



ventilating cubicles, racks and cabinets housing electronic equipment.

Mounted in an open enclosure, the unit is equipped with sleeve-type bearings and is powered by a 1/70-hp shaded pole electric motor operating on 115 volt, single-phase, 60/50 current cycle. At rated speed, the fan will deliver 450 cfm of air at zero static pressure and 100 cfm of air at approximately 0.22 in. wg static pressure.

American Radiator and Standard Sanitary Corporation, Industrial Div, Detroit 32, Mich.

HOT-WATER TEMPERING VALVE

With a temperature adjustment range from 110 to 180 F, the Tempo Type TMA Tempering or Mixing Valve is suitable for use on tankless heaters or storage tanks supplying domestic hot water. Offered in a ½-in. size with all connections for sweat copper, the valve utilizes a hermetically sealed thermostatic element and has a dial on top permitting easy setting of desired water temperature without the use of tools.

A. W. Cash Valve Manufacturing Corporation, P. O. Box 191, Decatur, Ill.

PIPE REPAIR CLAMP

Manufactured for pipe sizes from ½ in. through 8 in. and in widths of 3, 6, 9 and 12 in., the Heavy Duty Patchmaster clamp repairs pipe leaks either as an emergency measure or permanently. The lug design allows the clamp to conform to the contour of the pipe under high torque without biting into it. Able to withstand high clamping pressures without extruding, the Buna N pad is adaptable to oil, gas, water and steam.

Aeroquipt Corporation, Marman Div, 11214 Exposition Blvd., Los Angeles 64, Calif.

ELECTRONIC AIR CLEANER

Designed specifically for installation in homes having counter-flow or vertical-up-flow furnaces, this model is suited for houses having no basements and where utility space is limited. Since it has neither a built-in washing system nor a drain, its installation possibilities are virtually limitless. The ionizing-collecting cell is easily removed from the cabinet for cleaning.

Two Universal Line models with capacities of 800 to 1000 and 1200 to 1500 cfm are for installation with all forced air furnaces from 80,000 to 150,000 Btu input.

Electro-air Cleaner Company, Inc., Dept. UL, Olivia and Sproul Sts., McKees Rocks, Pa.

4 AND 6 CU FT REFRIGERATORS

Using no more than 180 to 220 watt, Pixie four and six cu ft refrigerators are designed for use in small apartments, motels, offices and trailers, and incorporate features of larger models.

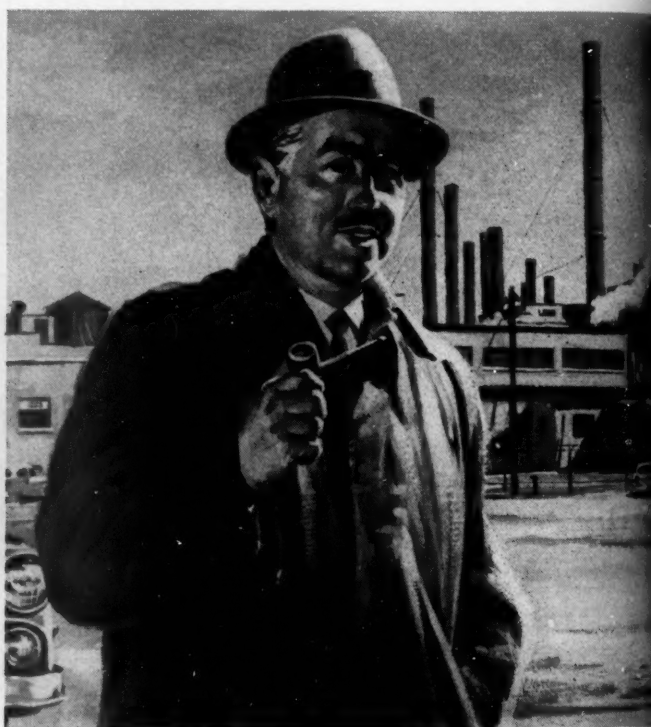
Norco, Inc., 5111 W. Washington Blvd., Los Angeles 16, Calif.

NEGATIVE PRESSURE SWITCH

When installed on the suction side of a system, this switch interlocks the ventilating equipment, fuel supply and safety devices, providing protection in applications where low suction will create a hazard. Capacity is 4

(*Continued on page* 102)

COPPER



"COPPER TUBE... for a job that will last." *says the contractor*

For refrigeration and air conditioning installations that go in smoothly, efficiently, profitably... that provide years of trouble-free service without "call backs" and complaints... you can put *your* confidence in copper tube!

"COPPER TUBE... nothing else is just as good" *says the manufacturer*

Copper tube is easier to work with! It can be formed, bent, brazed, soldered, welded and flared equally well in the plant and on the job. In addition, it provides an attractive looking installation that adds quality to the product.

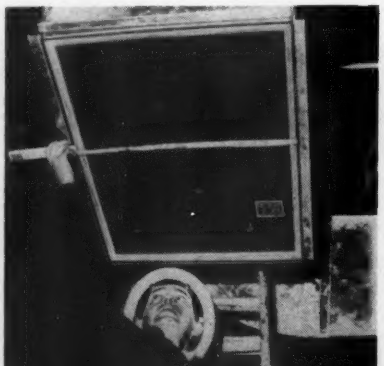
Specify Copper Tube for Air Conditioning

(*Continued from page* 99)

amp at 115 and 2 amp at 230 volt, ac or dc. Having a positive "on" and "off" indicator showing the position of the switch at all times, the Lo-Vac Unit closes at 0.21 in. and opens at 0.10 in. of water; maximum setting is from 0.30 to 0.15 in. of water.
Gas Appliance Specialties Company, Inc., 32-37 56th St., Woodside, N. Y.

GLASS FIBER DUCT LINER

Now available with a gray fire-resistant coating, Ultralite, this manufacturer's glass fiber duct liner, carries a



flame spread classification of less than 25 and is cited as conforming with requirements of various fire codes.
Gustin-Bacon Manufacturing Company, 210 West 10th St., Kansas City, Mo.

WATER DIVERTING VALVE

Independent unit temperature control plus self-contained automatic summer-winter switch over are advantages cited for Vectrole, a diverting valve control system for hot and cold water circuits. Consisting of three components, all available separately, the unit is powered from the air handling unit fan leads, is flare-fitted and may be serviced completely without removal from water lines.
Chatleff Valve and Manufacturing Company, P. O. Box 996, Austin, Tex.

TWO-STAGE SWITCH

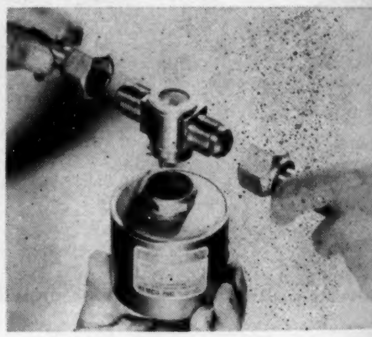
Combining a two level stage switching action to provide an electrical switch mechanism mounted over a pneumatic switch mechanism, Model J-1 S-1 VP Tandem Head makes possible such applications as air-operated level control with overriding electric action at high level for high-level alarm or shutdown, air-operated level control with underriding electrical action at low level for low-level alarm or shutdown, electric control of level with a pneumatic high-level alarm or shutdown, and electrical control with underriding pneumatic action for low-level alarm or shutdown.
Magnetrol, Inc., 2110 S. Marshall Blvd., Chicago 23, Ill.

LAMINATED POLYESTER FILM

Presently being used as a U-shaped slot liner in hermetic motors, where resistance to Refrigerant-22 and severe mechanical abuse during assembly are required, Lamicoid LPF-1 consists of two plys of 0.010 in. polyester film laminated with a bonding agent. The resultant laminate is cited as having excellent flexibility and mechanical strength, good electric properties, high heat resistance and resistance to delamination.
Minnesota Mining and Manufacturing Company, Mica Insulator Div, Schenectady 1, N. Y.

FILTER-DRIERS

Adapter type Molecular Sieve Filter-Driers and Moisture-Liquid Indicator Adapter Fittings, when combined in use, offer replacement of the former without disturbing line connections. Installed permanently in the refrigerant line, the adapter fitting enables fast installation and replacement of the filter-drier, which screws onto the bottom of it, as shown. Both units are



approved by Underwriters' Laboratories for 2500 psi minimum bursting pressure.

Cited as trapping particles as small as ten micron, with negligible pressure drop, the filter-driers also feature acid removal, keeping acid level below corrosion limits. The Molecular Sieves remove and retain up to 20% of their own weight in water at 140 F or higher, holding moisture concentrations to ten parts per million or less.
Remco, Inc., Zelienople, Pa.

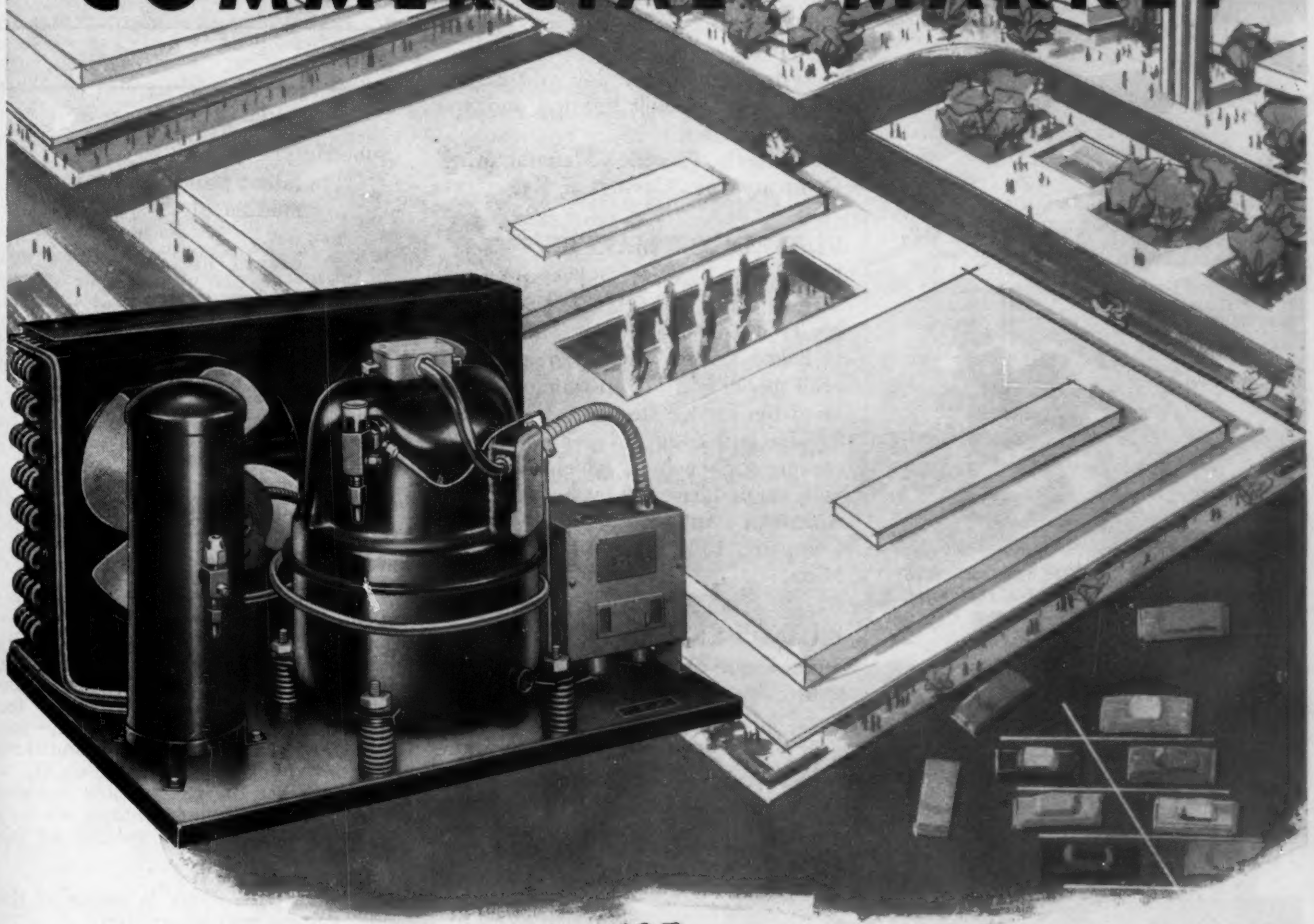
OIL BURNER CONTROL

For reducing unnecessary heating system lock-outs caused by rapid thermostat cycling and slow heat build-up in the stack, Type 611-1 is a constant ignition oil burner control using an

Tecumseh engineering VISION

37 MILLION COMPRESSORS IN THE FIELD

now offers full coverage of the COMMERCIAL MARKET



WITH 127 HERMETIC CONDENSING UNITS.

With the introduction of the new 2 and 3 H.P. low temperature hermetic units, the Tecumseh line is now complete for all normal commercial applications. Within the space of a few years time, this line has grown from only 9 units to today's blanket coverage.

Hermetic units now cover the range from 1/8 to 5 H.P. with variations for condenser cooling (air, water, and air-water), refrigerant, back pressure, torque and electrical characteristics. Every model has built-in sales features which are important to every user of refrigeration equipment.

This hermetic line is further backed up with 86 replacement compressors together with a full range of conventional equipment. Now you can sell matching Tecumseh equipment for all commercial applications. Investigate the outstanding advantages of going Tecumseh all the way!

TECUMSEH

The Leader Serving Leaders in the Air Conditioning and Refrigeration Industries

TECUMSEH PRODUCTS COMPANY

MARION, OHIO — TECUMSEH, MICHIGAN

FOREIGN OPERATIONS DIV: P. O. Box 2280, 24530 Michigan Ave., W. Dearborn, Michigan

CANADA: Tecumseh Products of Canada Limited, 1667 Dundas St., London, Ontario.

electric circuit that by-passes the effects of rapid cycling. Major change in the device is a shortened time cycle that removes the control's safety switch from the circuit in approximately 30 sec if flame is established.
White-Rodgers Company, 1209 Cass Ave., St. Louis 6, Mo.

MAIN SERVICE EQUIPMENT

Controlling all circuits, the 100 amp main Renu Fuse device has four fusible units which may be used for dryers, ranges, water heaters, air conditioners, etc., and may be had in eight, twelve or sixteen plug fuse circuits to be used for 240 volt heater circuits or 125 volt lighting and appliance circuits.
Wadsworth Electric Manufacturing Company, Inc., Covington, Ky.

FLAKED ICE MAKERS

Making, delivering and storing ice automatically, Model A-12 has a capacity of 120 lb, Model A-36 360 lb of flaked ice per day. Standard with each model is a 170 lb stainless steel-lined bin for ice storage. A-12 has a 1/5-hp and A-36 a 1/3-hp hermetic compressor, each of which is a 115 volt, single-phase, 60 cycle unit.
American Automatic Ice Machine Company, 1501 Park Ave., Faribault, Minn.

BACK PRESSURE REGULATOR

Suitable for pressures to 250 psi wsp and temperatures to 500 F, this pilot-operated regulator is available in ½- to 2-in. sizes from stock with screwed ends and 2½- to 3-in. sizes from stock with flanged ends. Used in industry where accurate control, tight shut-off and minimum maintenance are required in back pressure regulation, they can control 50 to 6600 lb of steam per hr or 7 to 190 gpm of water. Sliding gate seats, self-cleaning and self-lapping to insure tight shut-off, slice across flow to give balanced action.

OPW-Jordan, 6013 Wiehe Rd., Cincinnati 13, Ohio.

FOUR-WAY REVERSING VALVE

Rapid shifting of this valve, which is designed to shift while the system is in operation with a 300 psi differential, is cited as shortening changeover time, thereby shortening the defrost cycle. Made of cast iron and steel, the slide being of cast iron and the outer shell steel, the valve is so constructed that it may be mounted in any position except with the pilot valve upside-down. It can be installed to shift to cooling or to heating on power failure for the "fail safe" position.
Alco Valve Company, 865 Kingsland Ave., St. Louis 5, Mo.

IMPELLERS

(Continued from page 37)

Subscripts:

- f_g = latent heat
- $_m$ = meridional
- $_n$ = net
- $_o$ = overall, or input values
- $_p$ = polytropic
- $_s$ = static
- $_0$ = compressor inlet stagnation conditions
- $_1$ = impeller inlet (inside blades)
- $_2$ = impeller tip (inside blades)

APPENDIX I

Relationship between inlet volume flow and impeller tip flow coefficient

The Inlet Volume Flow, Q_o, usually expressed in cubic feet per minute, is the normally measured flow quantity of centrifugal compressors (or else, the weight flow and specific volume are obtained from conditions in the evaporator, from which $Q_o = w v_o$). Since the weight flow remains a constant through the stage, we may write the impeller tip volume as follows:

$$Q_2 = w v_2$$

hence the tip volume in terms of the inlet volume may be written

$$Q_2 = Q_o \frac{v_2}{v_o}$$

Now, it can be shown that the volume ratio, v_o/v_2 or its inverse, the density ratio, ρ_2/ρ_o, relating inlet conditions to the static conditions at the tip of the impeller, is dependent upon the Specific Heat Ratio of the vapor (k), the impeller tip static head coefficient, and the Rotational Mach Number, in an adiabatic process, as follows:

$$\rho_2/\rho_o = [1 + (k - 1)\ \mu_s\ M_o^2]^{\frac{1}{k-1}} \qquad (7)$$

Since most well designed impellers operate at notably high efficiency (sometimes reaching well above 90%), the process through this portion of the stage is quite nearly adiabatic. Also, a close approach to the impeller tip static head coefficient may be obtained by multiplying the useful overall stage head coefficient, μ_p, by a factor known as the "theoretical degree of reaction," R_s, which is an estimate of the amount or percent of the stage

head realized as static rise in the impeller. As shown in Fig. 3, this quantity is generally high for backward swept impeller blading, and is relatively low for radial bladed impellers. Including these quantities in the equation for density ratio, we obtain the following:

$$\rho_2/\rho_o = [1 + (k - 1)\ \mu_p\ R_s\ M_o{}^2]^{\frac{1}{k-1}} \quad (8)$$

We may now express the impeller tip volume as follows:

$$Q_2 = \frac{Q_o}{[1 + (k - 1)\ \mu_p\ R_s\ M_o{}^2]^{\frac{1}{k-1}}}$$

Also, in terms of the dimensions of the impeller and its outlet radial velocity, we find that the tip volume is also equal to:

$$Q_2 = 60\ c_{m2}\ A_2 \quad (9)$$

where A_2 is taken as the net tip area, or

$$A_2 = \frac{\pi\ d_2\ b_2}{144} - \frac{Z\ b_2\ e}{144 \sin \beta_2} \quad (10)$$

The second of these terms is the amount of the total area taken up by the blades, and for a great majority of impellers, this amounts to only 5 to 8% of the total projected area. On the average, we could write:

$$A_2 \cong \frac{0.93\ \pi\ d_2\ b_2}{144}, \text{ or simply } \frac{d_2\ b_2}{49.3} \quad (11)$$

In any event, we may write the following:

$$Q_2 = 60\ c_{m2}\ A_2 = \frac{Q_o}{[1 + (k - 1)\ \mu_p\ R_s\ M_o{}^2]^{\frac{1}{k-1}}}$$

or

$$c_{m2} = \frac{Q_o}{60\ A_2\ [1 + (k - 1)\ \mu_p\ R_s\ M_o{}^2]^{\frac{1}{k-1}}}$$

and

$$\phi_2 = \frac{c_{m2}}{u_2} = \frac{Q_o}{60\ u_2\ A_2\ [1 + (k - 1)\ \mu_p\ R_s\ M_o{}^2]^{\frac{1}{k-1}}}$$

which is the quantity and form in which we are interested. (Equation 12).

APPENDIX II

Sample calculations

Example: Using the data given in Figs. 3, 4 and 7 and Table II of this paper, determine the dimensions, speeds and relative power requirements for 300 ton Refrigerant-113 Centrifugal Refrigeration Compressors as follows:

(a) Single Stage with Radial Blades, at a maximum inlet relative Mach Number of 0.90.

(b) Two Stage (with economizer), with backward swept blading (just sufficient to place the design on the "envelope of current experience"), at a maximum inlet relative Mach Number of 0.90.

Solution for (a) Single Stage with Radial Blades: For the cycle

shown in Fig. 5 of the paper, a Refrigerant-113 compressor will be required to handle 14,560 cfm and produce a polytropic head of about 7190 ft. (Table II.) At an optimum inducer tip flow angle of about 32°, and an inlet relative Mach Number of 0.90, the impeller blade relative velocity = 0.90 × 372 (sound velocity) = 334 ft/sec, the inlet meridional velocity, $c_{m1} \cong 334 \times \sin 32° = 175$ ft/sec and the inducer tip peripheral speed = $334 \times \cos 32° \cong 285$ ft/sec, for no inlet prerotation. The net inlet area required for this design is

$$A_1 = \frac{14{,}560}{60 \times 175} = 1.387 \text{ sq ft}$$

(c_{m1} assumed uniform)

Allowing a free area factor of 0.825, accounting for the impeller hub and blade blockage effects, and for some expansion of the vapor from the evaporator to the impeller inlet, the required total impeller eye area becomes:

$$A_e = \frac{1.387 \times 144}{0.825} = 242 \text{ sq in.}$$

which results in an impeller eye diam of about 17.50 in. This dimension. taken with the optimum peripheral speed required at this diam, $u_1 = 285$ ft/sec, establishes the rotational speed in rpm:

$$N = \frac{285 \times 12 \times 60}{\pi \times 17.5} = 3730 \text{ rpm}$$

Referring now to Fig. 3, the polytropic head coefficient is a constant and equal to 0.69 for radial bladed impellers, $\beta_2 = 90°$ regardless of the Flow Coefficient. We may therefore determine the peripheral speed of the impeller from equation (1) of the paper as follows:

$$0.69 = \frac{32.17 \times 7190}{u_2^2}, \text{ or}$$

$$u_2 = 579 \text{ ft/sec}$$

which establishes the Rotational Mach Number $M_o = u_2/a_o = 579/372 = 1.554$, and the impeller tip diameter:

$$d_2 = \frac{579 \times 12 \times 60}{\pi \times 3730} \cong 35.50 \text{ in.}$$

The Impeller Tip Flow Coefficient $\phi_2 = c_{m2}/u_2 = c_{m1}/u_2$ for this radial bladed design since we should prescribe $K = c_{m2}/c_{m1}$ = no less than 1.00 according to Fig. 7. Thus, the Impeller Tip Flow Coefficient becomes

$$\phi_2 = \frac{175}{579} \cong 0.303,$$

which is certainly within the range of experience, according to Fig. 3. At this Flow Coefficient, and a Rotational Mach Number of 1.554, Fig. 3 indicates a relative polytropic efficiency of about .938 for Refrigerant-12. A Reynolds Number correction for Refrigerant-113 may be obtained from Fig. 4 by entering the chart on the bottom at a value of .938, then proceeding upward to the Refrigerant-113 line, and finally reading the corresponding relative polytropic efficiency of .893, at the left hand side of this figure. A relative gas horsepower may now be calculated as follows:

$$\text{Ghp} = \frac{\text{lb/min/ton} \times \text{ton} \times \text{head}}{33000 \times \text{rel. polytropic efficiency}}$$

$$\text{Ghp} = \frac{3.715 \times 300 \times 7190}{33000 \times .893} = 272.5 \text{ hp}$$

Solution for (b) Two Stage (Economizing) with Backward Swept Blades: This solution proceeds in a manner similar to the previous solution, except here we find a point on the "envelope of experience" in Fig. 3, where we can simultaneously satisfy both of the following conditions:

An inlet relative Mach Number of 0.90, with an optimum flow angle of about 32°, and

The maximum allowable deceleration rates given in Fig. 7.

This may be done quickly by trial and error, with the result that an impeller tip blade angle of 80° is found to be the maximum permissible. The (Fig. 3) impeller tip flow coefficient is 0.395 and the polytropic head coefficient is 0.64. This time, the required impeller tip speed is:

$$u_2 = \sqrt{\frac{32.17 \times 3760}{0.64}} = 435 \text{ ft/sec},$$

and the Rotational Mach Number is, $M_o = 435/372 = 1.169$.

With $c_{m2}/u_2 = 0.395$, $c_{m2} \cong 172$ ft/sec. Now Fig. 7 at an impeller tip blade angle of 80° indicates an allowable value of $c_{m2}/c_{m1} \cong 0.983$. Therefore, $c_{m1} = 175$ ft/sec (the same as for the single stage inlet—thus we have the same inlet velocity triangle, and inlet relative Mach Number as before). This time, the net inlet area becomes:

$$A_1 = \frac{12800}{60 \times 175} = 1.220 \text{ sq ft}$$

Again, with the same free area factor, the eye area becomes 213 sq in. and the eye diam is about 16.50 in.

The rotational speed in this case becomes:

$$N = \frac{285 \times 12 \times 60}{\pi \times 16.5} = 3960 \text{ rpm},$$

and we may also calculate the impeller tip diameter:

$$d_2 = \frac{435 \times 12 \times 60}{\pi \times 3960} = 25.20 \text{ in.}$$

Referring again to Fig. 3 at $\phi_2 = 0.395$ and $u_2/a_o = 1.169$, the relative polytropic efficiency, if Refrigerant-12 were the vapor, would be approximately .987. With the correction for Reynolds Number applied in this case, we may expect the relative polytropic efficiency of Refrigerant-113 to be about .950.

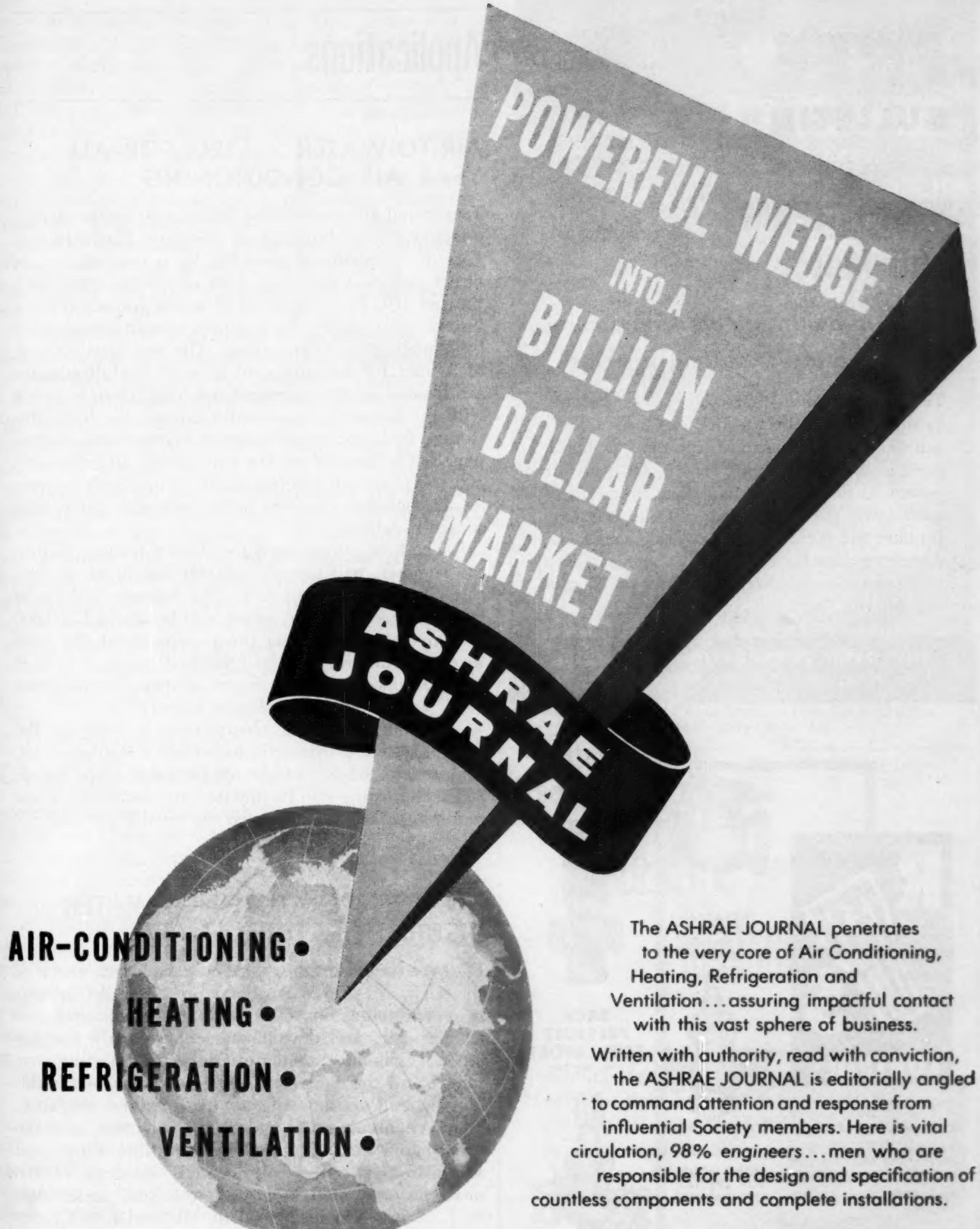
The corresponding relative gas horsepower for this stage (alone) can be calculated as follows:

$$\text{Ghp} = \frac{w \times H_p}{33000 \times \eta_p} = \frac{3.26 \times 300 \times 3760}{33000 \times .950} = 116.3 \text{ hp}$$

however, this cannot be compared directly with the overall hp of the single stage unit of Solution (a).

In order to arrive at more comparable figures (without going through an

(*Continued on page* 114)



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Applications

AIR-TO-WATER SYSTEM FOR ALL YEAR AIR CONDITIONING

Year-round air conditioning in the new System Headquarters Office Building of Western Massachusetts Electric Company is provided by a combination of direct radiation and dual duct air distribution, with the hot (105 F) or cold (45 F) water furnished by a 2-stage air-to-water heat pump system, engineered by Worthington Corporation. The two compressors, of 30 and 100 hp, and a 20 in. x 12 ft chiller-heater are located in the basement machine room, together with the hot and chilled-water pumps, the hot-water storage tank and supplementary heaters, and control panels. On the roof are the two outside air coils, each 100 sq ft, and circulating 52,000 cfm, which operate as air cooled condensers in the summer and a heat source in winter.

The heat pump, sized for 100-ton cooling load in the summer, will provide 829,000 Btu/hr at an outside air temperature of 35 F. The balance of the heat load at 0 F outside air, which will be about 1,200,000 Btu/hr in excess of heat pump capacity at the most severe conditions, will be furnished by an 8 x 23 ft storage tank in which a water temperature of about 200 F is maintained by electric heaters.

Although seasonal change-over is manual, the control system provides for automatic 2-staging of the compressors when outside temperature drops below 25 F. Defrosting will be initiated automatically, when required by frost build-up on the outside air coils.

HIGH TEMPERATURE HOT WATER DISTRICT HEATING AT AFA

Distance between the extreme developed areas of the U. S. Air Force Academy in Colorado Springs is approximately six miles; included in this area are: an academic and dormitory section, family housing areas, community center quartering supporting personnel and the Base Exchange, miscellaneous buildings making up staff officers' quarters and neighborhood schools, hospital and nurses' quarters, and the service and air field areas where utility shops and administrative offices are located. Analysis of the problems presented in heating this site, undertaken by J. O. Ross Engineering Div, Midland-Ross Corporation, indicated that two gas-fired high temperature hot water central plants and district heating systems should be provided for heating all buildings, and that individual family housing units should have their own gas-fired heat sources not connected to the district heating systems.

Situated in a central location with respect to load concentration, the main district heating system has a capacity of 400,000,000 Btu/hr and a maximum radius of distribution of 3 to 3½ mile, with the high-

est point in the distribution system 250 ft above and the lowest 250 ft below the boiler house floor; a separate 60,000,000 Btu/hr plant serves the airfield and service areas. Hot water is generated at 454 F and 440 psia. Steam pressurization is used, boilers being of the Lamont type. Circulation through them is maintained by a separate group of pumps.

The high level zone, supplying the academic area and the hospital, consists of four pumps, three of which are in simultaneous operation with the fourth on standby. At a maximum motor speed of 3550 rpm, 1300 gpm each are pumped, with a 335 ft total dynamic head, the total supply rate in this zone therefore being 3900 gpm.

Three pumps, with a maximum of two in simultaneous service and one on standby, comprise the low level zone supplying the Community Center and Staff Officers' Quarters. At a maximum speed of 3550 rpm, each pump delivers 375 gpm at 430 ft total dynamic head, for a maximum supply rate to this zone of 750 gpm.

In these systems, supply temperatures are maintained constant, and the varying flow rate reflects the heat absorbed in the users. To compensate for flow variations, the wound round motors have six speeds, the speed of all pumps in one group being set simultaneously by a common speed selector. The multiple pump arrangement allows further flow adjustment through adding to or subtracting from the number of pumps in operation.

NEW COMPRESSORS FOR MEAT PACKER

Recently installed in the South St. Joseph, Mo., plant of Swift and Company were three new compressors, two on ammonia refrigeration service and the third as a plant air compressor. Replacing a single refrigeration unit, the new machines are steam-driven Ingersoll-Rand Type XPV compressors, with the air unit working at 100 psig and the two refrigerant units supplying an average of 200 psig pressure. Two of the units were adequately housed in the area occupied by the predecessor compressor. The ammonia compressor has five-sixths the capacity of the old unit, but weighs less than half as much and occupies less than half the space.

Advantages cited for the new system, in addition to savings in floor-space, are lowering of maintenance costs, reduction in total down-time by over 90%, more economical operation because of better steam rate, and ability of the compressors to operate for long periods at full or nearly-full loads.

HEATING PLANT AREA TOTALS 7840 SQ FT

Designed as the first increment of a possible complete Base installation, the heating plant and related facilities of Camp Pendleton Marine Corps Base, San Diego, Calif., now consist of approximately 7840 sq ft total floor area. Installed because of its inherently

higher efficiency over either a central steam plant or individual battalion heating systems was a high temperature hot water plant and distribution system, with converting equipment in each of the buildings of the Camp requiring heating and domestic hot water.

Generators for the system are two Lamont type units with forced circulation provided by individual unit circulating pumps. These pumps are single-stage, motor-driven, steel-cased centrifugal units with water-cooled bearings. Bearing cooling water is provided by blow-through type evaporative condensers. An automatic combustion control system, sensitive to fluctuating pressures and temperatures, compensates for varying local weather and load conditions by proportioning of the fuel and combustion air.

Supply and return piping for the underground distribution system is standard-weight black seamless steel, made-up with standard weight butt welding fittings. Approximately 9980 ft in length, the system consists of 2½- to 8-in. piping, insulated with granulated Gilsonite cured at a temperature of 400 F.

LARGE HEAT PUMP SYSTEM

Heating and cooling for a new publishing house under construction in Huntington, Ind., will be provided by a large heat pump system. Three 500-hp water chilling machines, which will supply the needs of the 200,000 sq ft office and printing plant, are being constructed by Carrier Corporation. In lieu of pistons, the system uses the rotation action of impeller wheels spinning at high speeds in centrifugal refrigerating compressors, cited as providing larger capacities with fewer units and less maintenance.

NEW ORLEANS STREET AIR CONDITIONED

Intended as an air conditioning equipment promotion operation, the 900 block of Canal Street in New Orleans was provided with cooled air for eight hours on a July day. Each producing 18,000 cfm of dehumidified, filtered air 25 to 30 deg below the surrounding temperature, large portable ice air conditioning units manufactured by Ready Cool, Inc., were lined along the street. Plastics ducts leading from the units stretched into the air to force chilled air onto the sidewalk.

HEAT PUMP SAVES SPACE

Originally laid out for a steam boiler heating system, with space for water chilling machines, the Congregational Christian Church of Fairfax County, Va., changed to air conditioning by the heat pump method at the time of construction. Ten five-ton heat pumps built by Carrier Corporation were installed, use of two-piece units with the compressors outdoors gaining 415 sq ft of usable area for the church, space which would have been occupied by the steam boiler, water chillers and cleaning area in front of the boiler.

Outdoor units are ranged along the wall of an interior court and nine of the ten indoor sections are compactly stacked in a central equipment room occupying 285 sq ft of floor space and only 24 ft of exterior wall.

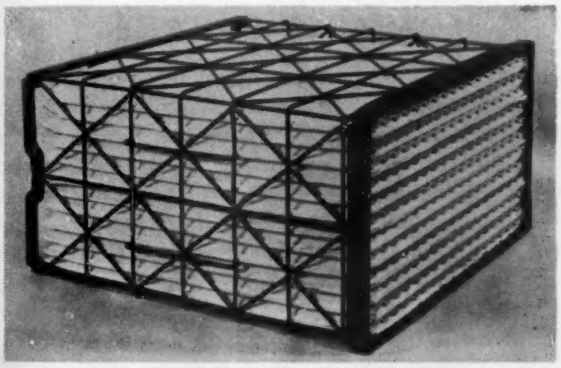
BASEBALL PARK GETS HEATED SEATS

Radiant heating for the San Francisco Giants' new Candlestick Park has been designed by Eagleson Engineers. Reserved seat patrons will be warmed by more than 35,000 ft of ¾-in. wrought iron pipe. Headers for the combination grid and coil type system will be placed in the aisles, with the distribution piping spaced on 18 to 48-in. centers and supported immediately below the precast concrete seats.

FELT USED IN ENGINE FILTER

Filter material in a heavy duty engine intake air cleaner had to meet exacting specifications: near 100% filtering efficiency in heavy dust conditions, ability to withstand oil, moisture, rot and mildew and to function in temperatures from –65 to 275 F.

Selected was a felt produced by Troy Blanket Mills under the trade name of Troyfelt. As protection against severe engine backfires, which often produce temperatures up to 1000 F, the exposed surfaces of



the filter were coated with a special compound. Used in this application was a felt of Dacron (polyester fiber); the felt is also produced in Orlon (acrylic fiber) or in special blends to suit specific purposes. Shown is the complete filter assembly, including basket framework and corrugated spacers.

PROBLEMS OF AIR DISTRIBUTION AND SPACE SHORTAGE OVERCOME

Installation of a 430 ton air conditioning plant in the Toronto office building of Bell Telephone Company presented two problems: first, an extremely high heat gain relative to the area in which it was generated had to be extracted, without the necessary air volume in circulation causing drafts on employees; and second, the existing building was crowded with equipment, leaving little space for the new unit. Especially designed by Carrier Corporation to offset heat generated by complex long distance electronic equipment, the system chosen was a steam absorption

refrigeration plant, rather than an electrically operated one, because it could be connected with the present boiler system. Installation of a completely new electrical service would have been necessitated by an electrical unit.

Because of the size and weight of the plant, location of it on the upper stories or roof of the building was impossible, and basement installation was prevented by lack of room. Finally, room was made for it on the first floor, requiring provision of additional supporting beams under the flooring to withstand the weight.

Three sections were to be serviced by the plant: an area of 1600 sq ft requiring 50 ton of refrigeration, another of 4800 sq ft requiring 93 ton of refrigeration and a third area housing long distance switching equipment requiring a lesser but still abnormal capacity. To cool the 1600 sq ft area by conventional methods would have meant blowing cold air into the occupied zone at the rate of 700 fpm. Solution to this was the building of special plenums above rows of panels, diverting the flow of air from the main air duct and forcing it between the panels. This almost completely eliminated any draft aspects of the system and proved an efficient method of cooling the bays of vacuum tubes that generated approximately 75,000 Btu per 40-ft row. Conventional duct distribution of the cool air proved successful in the other areas.

INFRA-RED WARMS GARAGE

Operating on the radiant heat principle, in a German-developed system incorporated into heating units by Perfection Industries Div of Hupp Corporation, infra-red heaters provide warmth for an Indiana service station. Six heaters, producing 144,000 Btu/hr, are hung overhead, and gas is consumed on a ceramic mat at a temperature of 1600 F. Most of the fuel's energy is converted by the heaters into infra-red rays that are readily absorbed by the surfaces they strike, maintaining a thermostatically-controlled 70 F.

BASEBALL PARK TO BE AIR-COOLED

In order to make the area more comfortable for baseball fans, 160 Gaffers and Sattler evaporative air cooler units, 15,000 cfm in size, will be installed in Chicago's Comiskey Park. Given proper weather conditions, the entire battery of coolers could lower the temperature in the stands from 10 to 15 deg. With unfavorable conditions, the units would still provide a breeze.

HP TOTALS 6000

Four 1500-hp centrifugal refrigeration water chilling machines and almost 5000 induction units, manufactured by Worthington Corporation, will provide air conditioning for the 8-story Central Intelligence Agency Building in Langley, Va. Four water chillers and self-contained units will be installed to handle special cooling problems, and two 2000 kw diesel

engine generator units will be on standby in event of central power station failure.

Nearly 100 centrifugal and vertical turbine type pumps, ranging in size from fractional to 250 hp, will be installed, some of which had to be designed for unusually high discharge pressures in order to lift large volumes of water from the central cooling plant to the top of the building.

UNIVERSITY CLASSROOM BUILDING TO HAVE HEAT-ACTIVATED CHILLER

Selected for the air conditioning system in the University of Hartford's new general classroom building is Carrier equipment which uses hot water to produce cooling.

Hot water at above normal temperatures supplied under pressure from a central plant activates the cooling equipment and is the source for winter heating and domestic hot water. The absorption chiller for the building works on the principle that evaporation of water produces cooling. In the process, water vapor is absorbed by a salt solution. Heat energy is used to boil off water absorbed by the solution in order to maintain a continuous process. Daily capacity of the machine is equal to cooling produced by the melting of 250 ton of ice, and will be energized by hot water at a temperature of 340 F. Future buildings in the system will be serviced by underground pipes carrying hot water from a boiler plant, and will each have its own absorption refrigerating machines.

IMPELLERS

(*Continued from page* 106)

analysis of the second stage for this machine), we have made the following assumptions:

The mean weight flow for the two stages is equal to the first stage weight flow plus one half of the economizer weight flow.

The overall two stage polytropic head is the same as the head required for the single stage machine.

The efficiency of the entire two stage machine is the same as the efficiency of the first stage alone.

It can be shown that these assumptions are nearly correct. In any event, the relative gas horsepower of the two stage unit, with economizer, in accordance with these assumptions, becomes:

$$\text{Ghp} = \frac{3.26 \times 1.063 \times 300 \times 7190}{33000 \times 0.950} = 239.5 \text{ hp}$$

The results of Solution (a) are given in Table III of the paper, while the results of Solution (b) are given in Table VI. The results for all of the other refrigerants and conditions shown in Tables III-VI were calculated in the same manner.

SOUND STANDARD

(*Continued from page* 49)

sweeps the room characteristics through a whole series. You get the effect of different frequencies. As a matter of fact, the room mode response is considerable. However, the practical aspects of the wings leave something to be desired, and also we still get some differences at low frequencies.

Another approach to the problem, which I personally think has a better chance of success, is to provide a diffusing element which will be placed around the walls and there has been a recent study, at a somewhat higher frequency level which gives us quite a little encouragement as to what can be done to improve diffusion by technique.

I do want to assure you that we do not want to rest with this situation. We can recommend and try to see what can be done to make a completely satisfactory room.

C. R. HIERS We have been talking about the size of a given room, what about the equipment to be tested in that room? For instance, one fan may be 2 in. diam and another 22 ft diam and use 2,000 hp. Should there be some perimeters laid down in relation to room size and specimen of test?

Mr. Ashley We have tried to show in Table I approximately what we feel to be the limiting sizes of equipment. I think it is already clear that the lower limiting size of the equipment is limited not by the size of the equipment, but by the other requirement of the room. But, on the other hand, it is obvious that a room of minimum size will probably not be able to take care of extremely large equipment.

In general, I think you can say that the perimeter of the equipment perhaps will be somewhere in the order of the largest dimension of the room. But, there is considerable latitude and we still have considerable to do. One of the things we hope to do in connection with further evaluating the basis for the standard is to run a round robin test of some equipment. And, admittedly, as we move forward, we are going to learn a lot more than we do now about the details of the method and the limit that we have to apply to it.

However, the information that we know has given us encouragement to feel that the standard as set forth is a realistic standard and any adjustment will be relatively minor to it.

Y. E. PRUSSLAT I wonder if you could elaborate a little more on your comment concerning the shortcomings of mixing vanes as a means of eliminating room geometry.

Mr. Ashley I am speaking of shortcomings other than mechanical. There are some serious mechanical problems. They have been working out in some laboratories where they have been used. One of the problems, in most of the laboratories that are using revolving vanes, they are using them for testing of wall samples or sound absorption or material of that sort. The vanes that are used are generally so large, they have to be large, that there is little space for the microphone or equipment to be tested. So that for our problem, entirely apart from the mechanical difficulties of the vane, they leave considerable to be desired.

FROST FACTOR

(*Continued from page* 77)

through the use of anti-sweat heater wires and non-conductive breakers. Where serious buildups occur, changes in design may be necessary.

CONCLUSIONS

As time passes and designs change many new developments will aid the engineer in combatting frost in spite of the increasing openness of cases. Until such time as frost can be eliminated entirely, it is necessary to keep as much moist air out of the cabinet as possible, defrost completely and frequently enough to insure an effective coil, and design the unit large enough to provide for frost accumulation. Field testing remains the safest way in which to determine the best possible method.